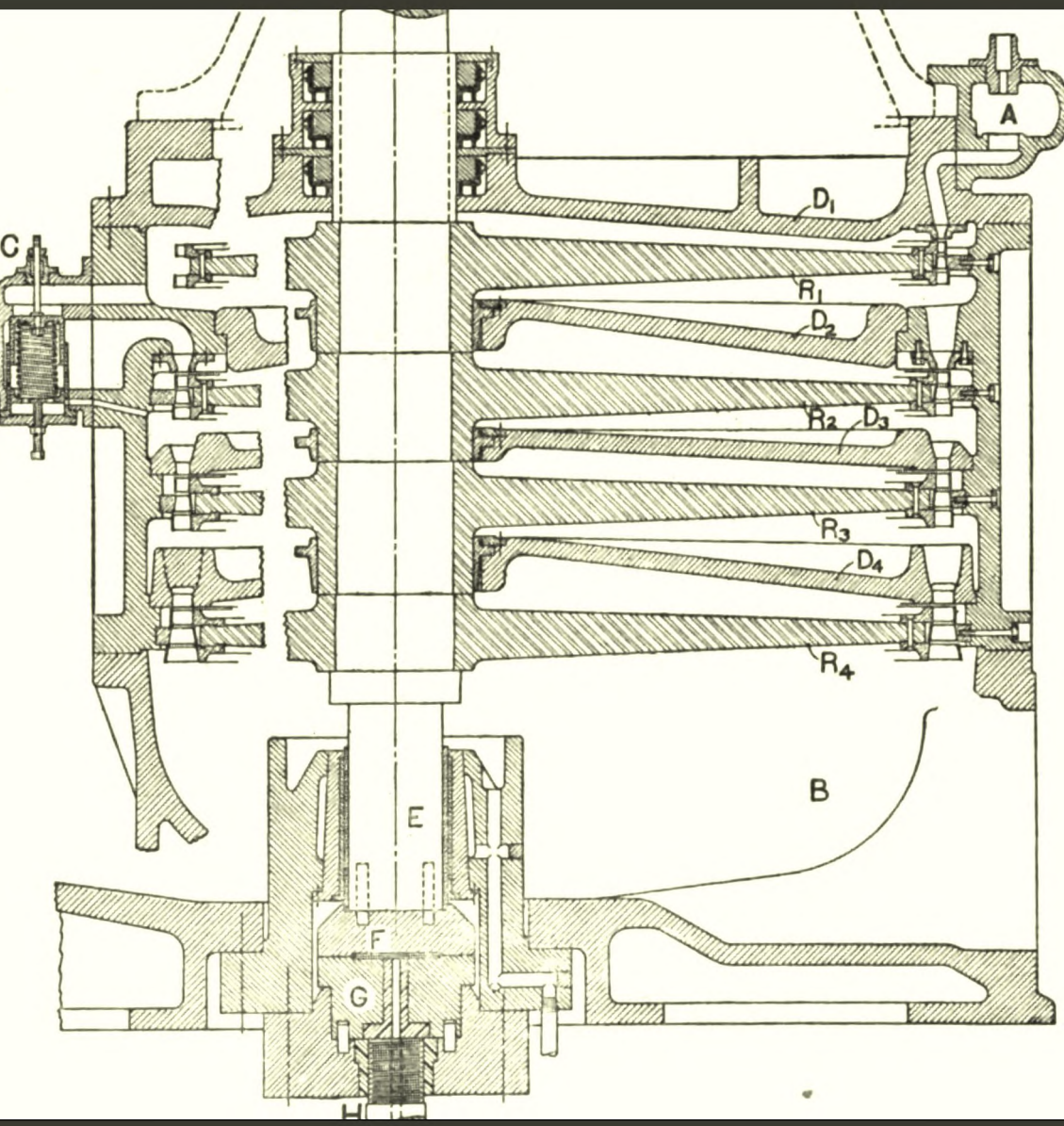

The steam engine and turbine

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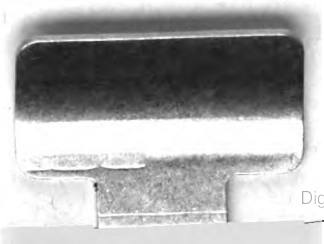

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THE STEAM ENGINE AND TURBINE

A TEXT-BOOK

FOR ENGINEERING COLLEGES

BY

ROBERT C. H. HECK, M.E.

Professor of Mechanical Engineering, Rutgers College

ILLUSTRATED

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PREFACE

In part this book is adapted from the writer's STEAM ENGINE AND OTHER STEAM MOTORS, in larger part it is rewritten or newly written. The textbook idea and the purpose of class-room use have continually been kept in mind. Mechanical form and manner of working are illustrated by selected, typical examples of construction; rational theory is built up, from fundamental concepts to the fully-developed ideal steam engine; and actual performance is studied and compared with the ideal, an especial effort being made to set forth clearly and logically the empirical knowledge which must fill the gap between them.

Viewing the steam plant as a whole, a line is drawn between the members that have to do with the generation and impartation of heat, and those concerned with its conversion into work through the agency of steam. In other words, the furnace and boiler, with their accessories, are taken to constitute a subject for treatment elsewhere, except that allusion is freely made to their functions. But on the side of the steam machine a comprehensive presentation is undertaken: to the writer it appears that the study of the piston engine and of the turbine can most effectively and profitably be combined in a single course.

It is assumed that the student approaches the subject with at least a general knowledge of the form and working of the steam plant, and with a good preparation in the elements of physics and of mechanics. All deductions along the latter lines begin, however, with basal facts or principles, so that the book shall be self-contained on that side. In the matter of thermodynamics, which is carried only so far as it is of immediate use and application, a special effort is made to develop concepts and ideas, not merely to build up a mathematical, abstract structure on a few axioms. An excess of mathematics is avoided, preference being largely given to graphical methods. Many numerical examples illustrate and enforce the text, emphasize the quantitative side of the subject, and will suggest problems for class-room use.

Certain omissions imply supplementation by other parts of the curriculum. Directions for making tests are left to the laboratory course, although the purpose of such tests, the quantities sought, and the

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manner of workup are fully indicated. Design goes no farther than proportioning of the parts that have to do with steam action, not extending into the field of machine design. Plant layout and economics, to a generally sufficient extent, can be covered by a few lectures, with visits of inspection. Engine or plant manipulation is hardly a matter to be learned from books.

A new steam table is presented, differing slightly from that of Marks and Davis, but founded on the same experimental data. For saturated steam, it was worked out before the latter appeared; in continuity and smoothness it is a little better, especially in the high range of pressure; while for superheated steam, the line of equal temperature, used in Table VII and described on page 86, is a new and most effective means of correlating data as to total heat.

As between engine and turbine, the former receives much the larger share of space, although the difference is less than may appear, since Chapters I to IV and the first part of Chapter VI apply to both. Of intention, however, the engine is given fuller treatment: the problems of its intimate behavior have been more fully worked out or solved by experience, so that they come within textbook scope; and a general text like this may aim to be fairly complete, within its field, for the engine, while very properly making frequent reference to large and special works on a growing subject, such as Stodola's *Steam Turbines*.

The arrangement of the book is intended to facilitate selection and omission, with §§ 22 to 25 most strongly indicated for briefer summarization in lecture. The section on condensers brings that rapidly-developing subject up to date, in good shape for the student.

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THE STEAM ENGINE AND TURBINE

CHAPTER I

A GENERAL VIEW OF THE SUBJECT

§ 1. The Steam-Power Plant

(*a*) A SIMPLE STEAM-ENGINE PLANT is outlined in Fig. 1, with the purpose of showing clearly what are the essential elements or organs of a complete apparatus for the generation of power by means of heat, derived from the combustion of fuel and utilized through the medium of steam. These organs are:

I The Boiler (including the furnace as well as the boiler proper), where the fuel is burned and the steam is generated.

II The Engine (whether piston engine, as here, or steam turbine), in which the expansive force of the steam is applied to the doing of useful work.

III The Condenser, which receives the used or exhaust steam and abstracts its heat, bringing it back to the initial state of water. Quite frequently the condenser is omitted from the plant, its function being taken by the atmosphere, into which the steam is then exhausted.

IV The Feed Pump, which returns to the boiler either the condensed steam or an equivalent amount of fresh water, thereby completing the cycle of operations.

The boiler and engine are naturally considered the principal members of the plant, while the condenser and feed pump, with the feed-water heater, come under the head of auxiliaries. The more important parts are named under Fig. 1, and this list of names is to be used in connection with the following description of the working of the plant.

(*b*) THE FUNCTION OF COMBUSTION. — In the operation of the steam generator two distinct sets of phenomena are involved, those of combustion and of heat transfer and evaporation. The essential condition for combustion is that a sufficient supply of air be continually brought into contact with the fuel. To secure this, there must be first

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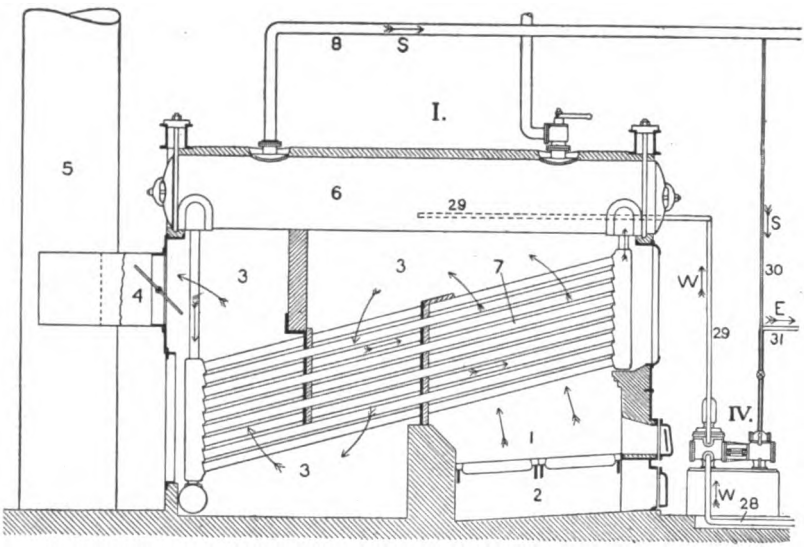

FIG. 1. — The Steam Plant. A. Boiler and Feed Pump.

I The Boiler (Water-tube type).

1. Grate and fire space.
2. Ash pit.
3. Hot-gas spaces.
4. Flue and damper.
5. Chimney.
6. Boiler shell or drum.
7. Tubes.
8. Steam pipe.

II The Engine (Corliss type).

9. Throttle valve.
10. Cylinder.
11. Engine frame.
12. Piston rod.
13. Crosshead.
14. Connecting rod.
15. Crank.
16. Fly wheel.
17. Governor.
18. Exhaust to condenser.
19. Exhaust to open air.

a suitable arrangement for holding the bed of fuel, so formed that air can pass through it, with provision made for introducing fresh fuel and removing the solid waste products; second, means for regulating the supply of air, both below and above the fire, and for producing and regulating the draft which draws or forces the air and the combustion gases through the fire and along the passages through or around the boiler; lastly, a sufficient space above the fire, in which combustible gases from the solid fuel can be completely burned before they are cooled below the ignition temperature by contact with the relatively cold surfaces of the boiler. In Fig. 1, these requirements are met by the grate, ash pit, and fire space, with the fire door and ash-pit door, both provided with air grids; and by the chimney and damper. With special fuels, liquid or gaseous, other arrangements take the place of the grate

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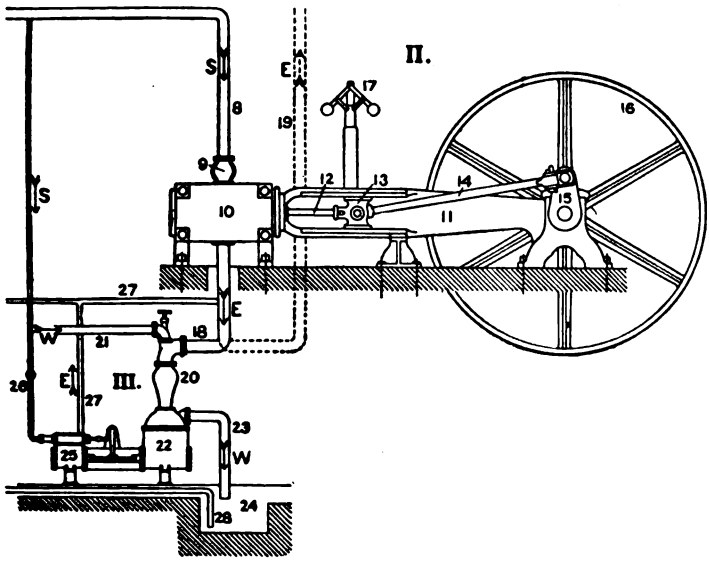

FIG. 1. — The Steam Plant. B. Engine and Condenser.

III Condenser and Pump (Jet or mixing type).

20. Condensing chamber.
21. Cold-water supply.
22. Pump cylinders.
23. Discharge pipe.
24. Hot well.
25. Steam cylinders.
26. Steam pipe to pump.
27. Exhaust pipe.

IV The Feed Pump (Separate, steam-driven type).

28. Suction pipe.
29. Feed pipe.
30. Steam pipe.
31. Exhaust pipe.

used with solid fuel. And various forced-draft appliances are frequently used to assist, or partially to replace, the chimney.

(*c*) THE FUNCTION OF EVAPORATION. — In order that the boiler may freely absorb the heat generated by combustion, it must have a large area of heating surface, so disposed that there will be a rapid flow of the hot gases over the outer side, and that the steam as formed will be able to escape freely and rapidly from the inner side. A small proportion of this surface is exposed to radiant heat from the solid fuel and from incandescent flame: this direct heating surface is far more effective in absorption than is that which receives heat only by contact and conduction from the hot gas. The current of gas is split up into narrow streams, and the body of water is likewise divided into small parts, so that there shall be only a slight depth of gas acting upon, and of water

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heated by, any particular portion of surface. Whether this intimate contact is secured by the water-tube arrangement of Fig. 1, or by the fire-tube arrangement of cylindrical boilers, is a matter of minor importance. Means for insuring a full circulation of the hot gases over the whole of the heating surface are shown in Fig. 1; and the boiler is so formed as to permit free internal circulation, whereby a current of mixed water and steam bubbles is continually rising through the front connecting tubes into the drum, where there is ample surface for the separation of the steam from the water.

(*d*) THE BOILER A SEPARATE SUBJECT. — The above general considerations are here stated in full because they are fundamental to an understanding of the thermal performance of the boiler, as a member of the steam plant. But the boiler is made in so great a variety of forms, and there are so many special matters involved in its design, construction, and management, that it properly forms a separate subject — together with all the appliances for handling and controlling steam, such as piping, valves, steam traps, separators, etc. No further description of the boiler or of its accessories will be given in this book; but a fair working knowledge of these parts of the plant is assumed, and reference to their functions will be freely made when the principles involved come under discussion.

(*e*) THE ENGINE. — Simple representative examples of both the piston engine and the turbine engine will be described in this chapter, as to form and operation. In general, the engine may be considered as a thermodynamic apparatus (involving relations between heat and work), and as a machine (involving motions and the action of forces): both these phases of the subject are to be fully developed in this treatise.

(*f*) CONDENSING THE EXHAUST STEAM. — The two ways of getting rid of the exhaust steam are indicated in Fig. 1. The simplest is, of course, open exhaust to the air; but the efficiency of the engine can be increased by condensing the steam at low temperature and in a consequent vacuum, using a pump to remove the water and maintain the vacuum. In the figure, the exhaust meets, in the chamber 20, a jet of cold water from the pipe 21, and is condensed by direct contact and mixing. The water from the condenser, moderately warm, is discharged to a tank called the hot well.

The difference here described marks the distinction between condensing and noncondensing engines or plants. A brief description of the several types of condensers will be found in Chapter XI.

(*g*) THE FEED PUMP AND FEED-WATER HEATER. — In the simplified plant in Fig. 1, the feed pump draws from the hot well an amount of

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water equal to the steam condensed, and forces it directly into the boiler, at hot-well temperature; the rest of the warm water runs to waste. In a fully developed, well-designed plant of this type, the exhaust from the pumps would not go into the main condenser, but into a feed-water heater, where perhaps all of its heat can be utilized in raising the temperature of the boiler feed. With open exhaust (engine noncondensing), and with water drawn from a cold supply, the feed-water heater is essential to economy, and should never be omitted from the plant — unless controlling conditions inhibit its use, as on the locomotive. The matter of heating the feed water is discussed, with reference to steam-plant efficiency, in Chapter VI, § 26 (*d*).

§ 2. Construction and Working of the Engine

(*a*) Representative Examples, of contrasting types of design, are shown in Figs. 2 and 3 and further detailed in Figs. 5 to 8. The first is of the short-stroke, compact and self-contained, high-speed type; the second has a relatively long stroke of piston, is of more drawn-out and open form, has the Corliss arrangement of valves and valve gear, and runs at a much lower speed of rotation.

The smaller engine is said to be self-contained because all of the parts, including cylinder and bearings, are carried by the bed and sub-base; as appears in Fig. 7, it is of the center-crank form, with two main bearings symmetrically located and with two wheels. Even if made with a side crank, after the manner of Fig. 8, this engine would still be self-contained to the extent of having the cast-iron sub-base extend out beneath the outer bearing. In the Corliss engine, the cylinder is so long and heavy that it must be fully supported by the foundation; and the outer or outboard bearing is independently carried, by a pier which extends up from the main base or body of the foundation. A lighter and more open type of frame is outlined in Fig. 1.

These are both what are called simple engines, because the steam does all its work in one cylinder. The compound engine receives steam into a smaller, high-pressure cylinder, and after a part of the work has been done the steam passes to a larger, low-pressure cylinder, and thence to the exhaust. In some lines of service, this scheme is extended to include three or four such successive steps or stages in pressure drop and expansion. See § 20, Chapter V.

(*b*) The Engine Mechanism, drawn in skeleton outline in Fig. 4, consists of three moving members, besides the fixed bed or frame. These are, the sliding piece made up of piston, piston rod, and crosshead, the connecting rod, and the rotating crank or shaft. The slide, reduced

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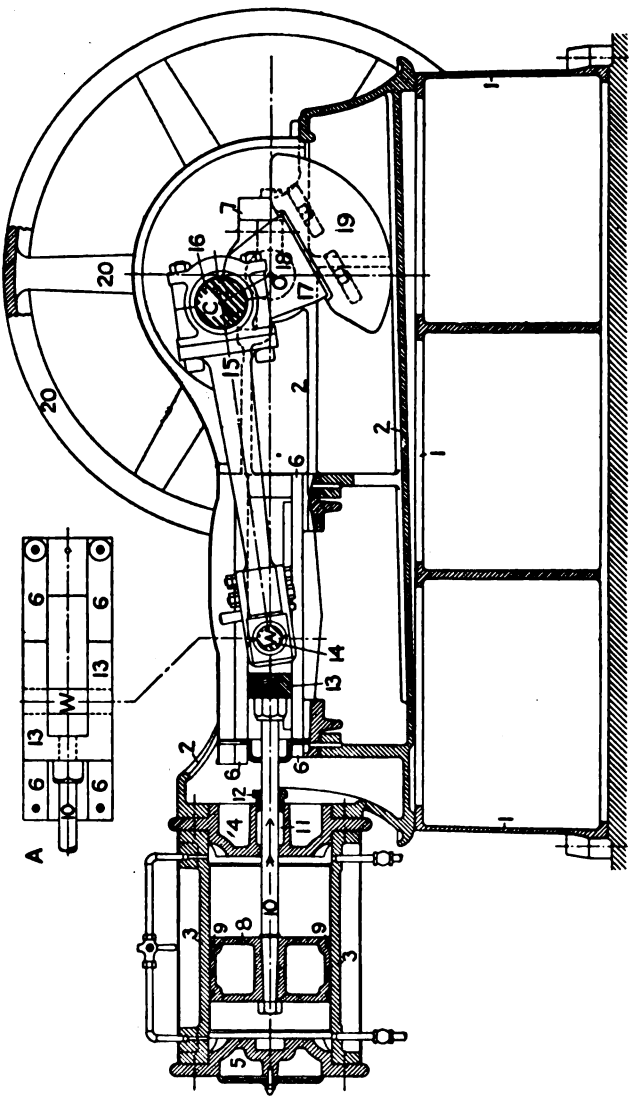

FIG. 2. — The Short-stroke, High-speed Engine; American-Ball design, 15 in. diameter by 14 in. stroke, to run at about 250 revolutions per minute. Scale 1 to 24. At A, top view of crosshead and guides. Cross sections in Figs. 5 and 7, and details in Figs. 202 and 208.

The following list of parts applies to Figs. 2 and 3:

1. Sub-base.
2. Frame, body, or bed.
3. Cylinder.
4. Front cylinder head.
5. Back cylinder head.
6. Guides for crosshead.
7. Main bearing.
8. Piston.
9. Packing rings.
10. Piston rod.
11. Stuffing box.
12. Stuffing-box gland.
13. Crosshead.
14. Wrist pin.
15. Connecting rod.
16. Crank pin.
17. Crank arm or disc.
18. Shaft.
19. Counterweight.
20. Fly wheel.

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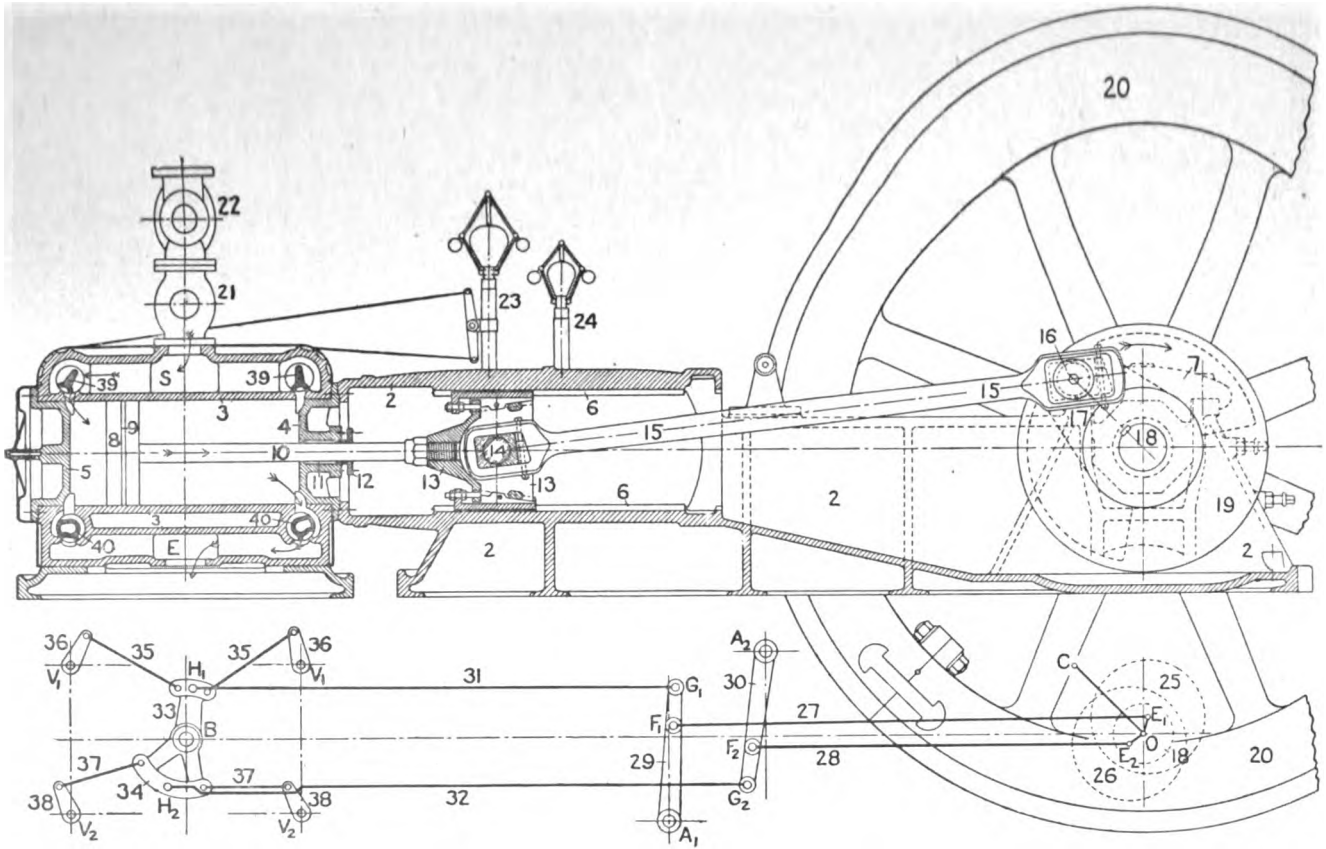

FIG. 3. — The Long-stroke, Corliss-type Engine; Murray-Corliss design, 26 in. diameter by 48 in. stroke, to run at 90 revolutions per minute. Scale 1 to 48. Cross sections in Figs. 6 and 8, details in Figs. 204, 206, 212, and 217.

List of parts shown in Fig. 3 only.

21. Main throttle valve.	24. Safety governor.	27, 28. Eccentric rods.	33, 34. Wrist plates.	39. Steam valves.
22. Safety-stop valve.	25. Steam eccentric.	29, 30. Rocker arms.	35, 37. Valve rods.	40. Exhaust valves.
23. Main governor.	26. Exhaust eccentric.	31, 32. Reach rods.	36, 38. Valve arms.	

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in Fig. 4 to a simple block around the wrist pin, is the work-receiving member, upon which acts the steam force P. Transmitted by the connecting rod to the crank pin as R (with some modification), this force turns the crank against the external load-resistance. The effect of the mechanism is, then, to change from a back-and-forth or reciprocating, straight-line motion of the piece which receives the driving force to a rotary motion of the piece which moves against the load

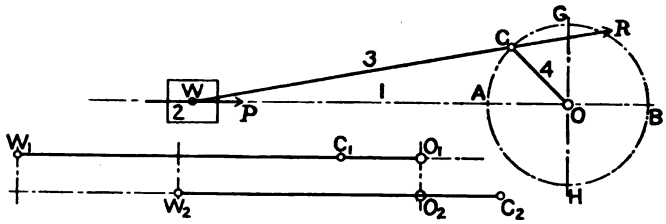

FIG. 4. — The Engine Mechanism, and Principal Driving Forces.

1. Frame.
2. Piston slide.
3. Connecting rod.
4. Crank.

W. Wrist pin.
C. Crank pin.
O. Shaft.

force. In some types of engines, however, the motion of the piston is applied directly to the useful resistance — as in pumps, compressors, blowing engines, and steam hammers.

It is evident from Fig. 4 that the turning effect of the force R will be relatively greater when the crank is near the vertical positions OG and OH, than when it is near OA and OB. When crank and rod are in line, as sketched below the main diagram, the piston being at one or the other extreme or limit of its stroke line or travel range, the engine is said to be on dead center; in these positions there can be no tendency to turn the crank, no matter what force along the stroke line or cylinder axis is exerted at W. The essential function of the fly wheel or balance wheel is to moderate the wide variations in turning effect, and to restrain within a very narrow range the corresponding fluctuations in the speed of the shaft. The action of the forces in the machine forms the subject of Chapter VII.

(*c*) THE PISTON SLIDE. — In Fig. 2 the piston is shown as a plain, thick disc, made hollow for lightness, but broad of face so as to have a liberal bearing surface where it slides within the cylinder. The piston of Fig. 3, of more complicated construction, is given in detail in Fig. 206. There is a loose fit between cylinder wall and piston, and the joint is made steam-tight by the packing rings, which are set into grooves cut in the piston rim, and are pressed outward either by their own elasticity or by light springs placed beneath them. The piston rod, securely

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fastened into piston and crosshead, passes out of the cylinder through a stuffing box, which consists of an annular space filled with a fibrous packing material, closely pressed into place around the rod so as to

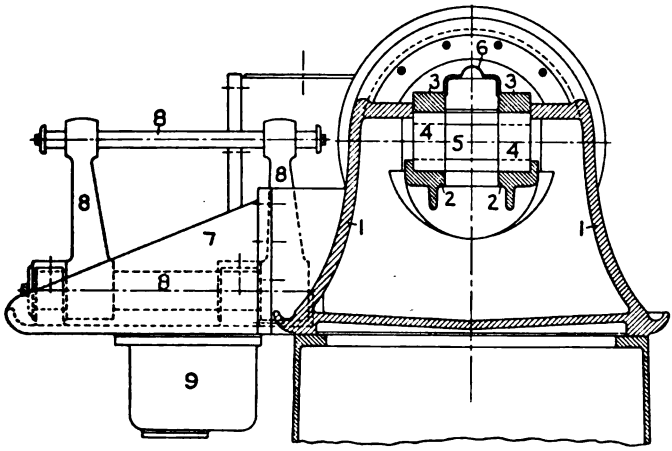

FIG. 5 — Engine in Fig. 2, Cross Section at Guides.

1. Engine bed.
2. Bottom guides.
3. Top guides.
4. Crosshead.
5. Wrist pin.
6. Oil guard.
7. Rocker bracket.
8. Rocker arm.
9. Oil tank.

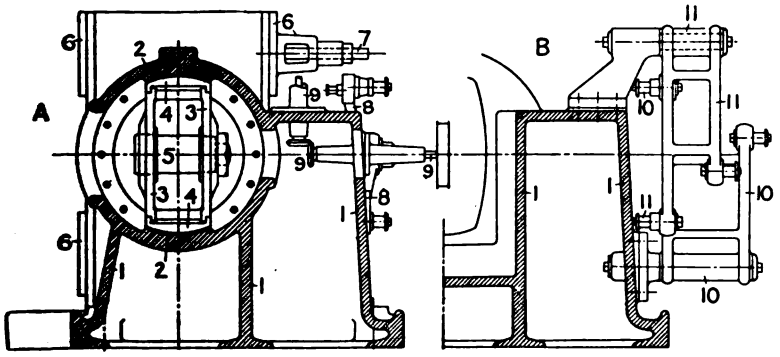

FIG. 6. — Engine in Fig. 3, Cross Sections, A at middle of guides, B in front of guides, both looking toward cylinder.

1. Engine bed.
2. Guides.
3. Crosshead.
4. Crosshead shoes.
5. Wrist pin.
6. Valve bonnets.
7. Valve stem.
8. Wrist plates.
9. Governor base and drive.
10. Steam rocker.
11. Exhaust rocker.

prevent leakage of steam. The crosshead carries the wrist pin, which forms the joint with the connecting rod, and guides this pin along its straight-line path. The crosshead of Fig. 2, as shown by view A and by Fig. 5, has its sliding surfaces on blocks which extend out from the

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sides like wings, and requires four guide bars; it is appropriately called the wing or four-bar type. In Figs. 3 and 6, the crosshead is of the box or trunk type, sliding between guides placed symmetrically above and below the axis; the rubbing surfaces are on separate shoes, with wedge adjustment to take up wear.

(*d*) THE CONNECTING ROD. — It is through the motion of this piece that the working force changes from a rectilinear to a circular path. Structurally, it consists of the shank or body and two heads which carry the bearings for wrist pin and crank pin. The possibility and

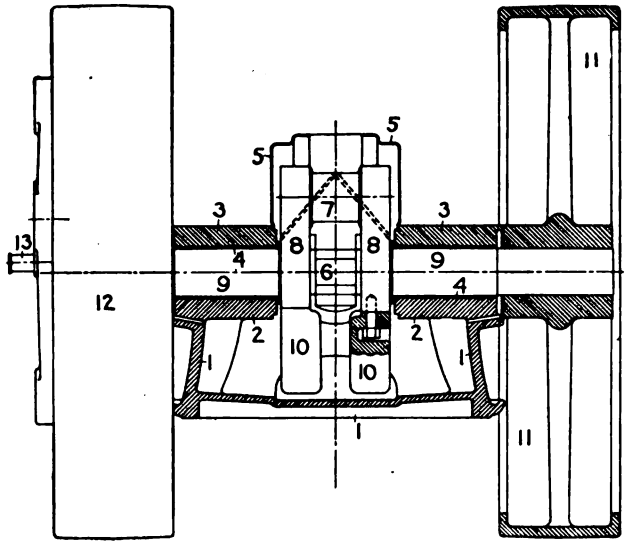

FIG. 7. — Engine in Fig. 2, Cross Section at Bearings.

1. Engine bed.
2. Main bearings.
3. Bearing caps.
4. Bearing shell, of babbitt metal.
5. Oil guard.
6. Wrist pin.
7. Crank pin.
8. Crank webs.
9. Shaft, journals.
10. Counterweights.
11. Plain wheel.
12. Governor wheel.
13. Eccentric pin.

method of adjusting these bearings is evident from the drawings. At the wrist pin in Fig. 2 there is a bolted strap end, at the crank pin a "marine" end; the rod in Fig. 3 has two solid ends. Being subject to a violent swinging motion in its plane of movement, the rod of the high-speed engine in Fig. 2 has its shank formed with a deep rectangular cross section; for low-speed service the rod body is round, as in Fig. 3.

Piston slide and connecting rod together constitute the reciprocating parts of the engine. At high speeds, the inertia forces due to their rapid acceleration in alternate directions have a very considerable effect upon the force action in the engine.

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(*e*) Crank Shaft and Wheels. — In Fig. 7 the shaft with inside crank is a solid forging, having the crank pin of the same diameter as the main journals, and with the counterweights bolted on, as shown also in Fig. 2. In Fig. 8 the end crank is built up, the pin is much smaller than the shaft, and the counterweight (see dotted outline in

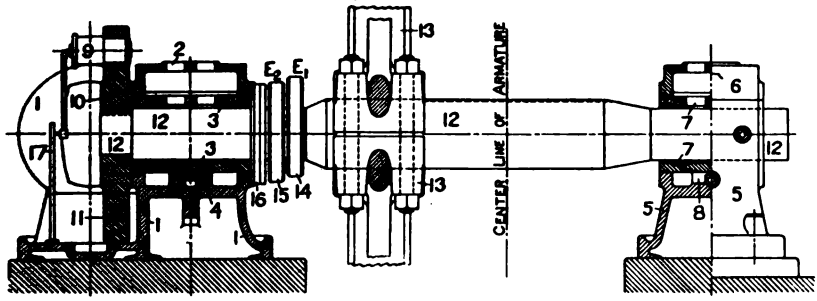

Fig. 8. — Engine in Fig. 3, Cross Section at Bearings.

1. Engine bed.
2. Main bearing cap.
3. Bearing shells.
4. Adjusting wedge.
5. Outboard bearing.
6. Bearing cap.
7. Bearing shells.
8. Adjusting wedge.
9. Crank pin.
10. Crank disc.
11. Counterweight.
12. Shaft.
13. Balance wheel.
14. Steam eccentric.
15. Exhaust eccentric.
16. Governor pulley.
17. Crank-pin oiler.

Fig. 3) is a part of the cast-iron crank disc; on account of its length and of the very heavy weights which it carries, this shaft is enlarged between the bearings.

The counterweight is put on to "balance" the engine; its centrifugal force acts against the inertia force of the reciprocating parts, and thus greatly diminishes the free or unbalanced force which tends to shake the engine on or with its foundation. See § 30 (*g*) and § 35.

Figs. 2 and 7 show belt-pulley wheels, of small diameter and cast all in one piece. In Figs. 3 and 8 the wheel (16 ft. in diameter) is made in halves, which are held together by heavy bolts at the hub and by I-shaped shrink bolts or links at the rim; the deep rectangular rim is of the balance-wheel type. The function of the fly wheel, to maintain uniformity of rotary speed, has already been alluded to; it is fully discussed in § 33.

(*f*) Engine Bed and Bearings. — In stationary engines, as here illustrated, the frame or body is usually a casting of massive external form, but of course made hollow, with ribs or webs to give the needed strength and stiffness. Transportation engines, locomotive and marine, have a lighter but more complicated structure, to which the name framework is more properly applicable. In the engine of Fig. 2 the bed, supplemented by light oil guards, forms a casing or inclosure

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about the working parts, to retain and collect all the oil that escapes from the various bearing surfaces or is splashed from the moving parts.

The main bearings in Fig. 7 are of unusually simple form, with thin shells of Babbitt metal set into place about the journal. In Fig. 8 (see also Fig. 217) there is vertical adjustment at both bearings, adjustment at one side of the main bearing for wear, and adjustment at both sides of the outboard bearing for alignment of the shaft.

(*g*) LUBRICATION. — The most obvious scheme of oil supply for the bearing surfaces of an engine is a set of sight-feed drip cups, on all the bearings, to be filled and regulated by hand. A more advanced method is to feed oil from a central tank or reservoir, by pipes leading to all the bearings, through individual sight-feed nozzles, at which the flow can be regulated. Both the engines here illustrated have small oil pumps, driven by the valve-gear rocker arms, which return the oil to the tank as it runs together after escaping from the bearings; in Fig. 5 the collecting tank from which the pump draws is shown, beneath the rocker bracket. In a large plant, a group of engines or turbines is often served from a single installation of this sort, with a separate oil pump, oil filters, and a central tank at the low and at the high level.

To get oil to the crank pin, without the possible need of stopping the engine to refill a cup that has run dry, is a problem which called for some ingenuity when it was first solved. In Fig. 7, typical of small self-contained engines, oil is fed through holes drilled in the solid crank shaft; it is collected from the inner ends of the main bearings by annular grooves in the outer faces of the crank webs, and is carried to the pin by centrifugal force. In Fig. 8 is outlined the device generally used on open, side-crank engines; an oil pipe in the form of a return crank runs from the crank pin inward to the shaft axis, and at this "fixed point" oil can easily be fed into the pipe.

The lubrication of the internal sliding surfaces of valve and piston is effected by using the steam as a vehicle to carry the oil. By means of a sight-feed cylinder lubricator, or of an oil pump operated by the valve gear, oil of suitable quality is slowly and continually fed into the steam pipe close to the steam chest. There it is broken up or "atomized," and is carried by the steam (or by moisture in the steam) to the rubbing surfaces. In many cases the oil is more directly fed upon the admission valves, but even then the steam has a good deal to do with its distribution over the valve surfaces, and must still be depended upon to carry oil to the surface of the cylinder.

(*h*) CYLINDER AND VALVE CHEST. — Having considered the form and working of the engine, as regards its machine action in receiving, transmitting, and delivering force, we come next to the question of how

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the primary working force is developed and how it varies; that is, to the matter of steam-force action and of steam distribution or the control of steam flow. Since all the operations involved are performed within the cylinder, this part of the engine is now more fully illustrated in Fig. 9, which is especially intended to show the form and arrangement of the valve, valve chamber, and steam passages. The simple slide-valve engine is taken as the basis of the description of valve action to be given presently, all consideration of the more complex Corliss gear being reserved for Chapter VIII. Fig. 9 does not belong to the engine

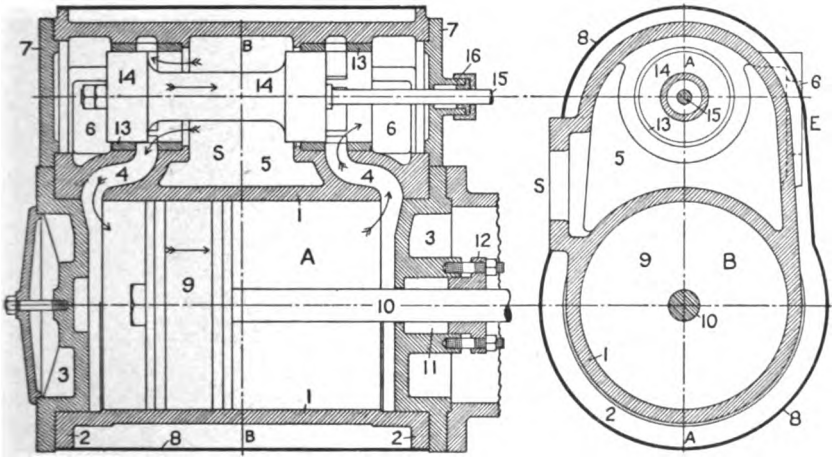

FIG. 9. — Cylinder with Piston Valve and Inside Admission; A, horizontal section along axes; B, cross section at mid-length.

1. Cylinder shell.
2. Cylinder flanges.
3. Cylinder heads, front and back.
4. Steam ports.
5. Steam space.
6. Exhaust spaces and outlets.
7. Steam-chest covers.
8. Cylinder sheathing.
9. Piston.
10. Piston rod.
11. Stuffing box.
12. Gland.
13. Valve seats.
14. Piston valve.
15. Valve rod.
16. Stuffing box.

in Fig. 2 — see Fig. 202 for the valve arrangement of the latter — but is of the same class; it has a piston valve, not quite so simple in form as the plain flat valve used in Figs. 13 and 14, but the same in effect. The central steam space, the steam ports or cylinder ports, and the exhaust spaces are clearly enough described by the drawing. The throttle valve would be placed just above the steam chest, as in Fig 3; and with the two exhaust outlets here shown a special Y pipe must be placed beneath the steam chest to make connection with the exhaust pipe proper.

Evidently, a short reciprocating movement of the slide valve, on its seat, over the ports, will accomplish the desired end of alternately

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admitting steam to, and allowing it to escape from, each end of the cylinder.

(*i*) THE INDICATOR AND ITS DIAGRAM. — The performance of the steam, controlled by and resultant from the action of the valve, is best shown by the steam diagram, of which a typical example is given in Fig. 11. This is drawn autographically by an instrument called the steam-engine indicator, illustrated in Fig. 10. The indicator measures the rapidly varying pressure in the engine cylinder, and records it in terms of piston position; the diagram is on the rectangular coördinate

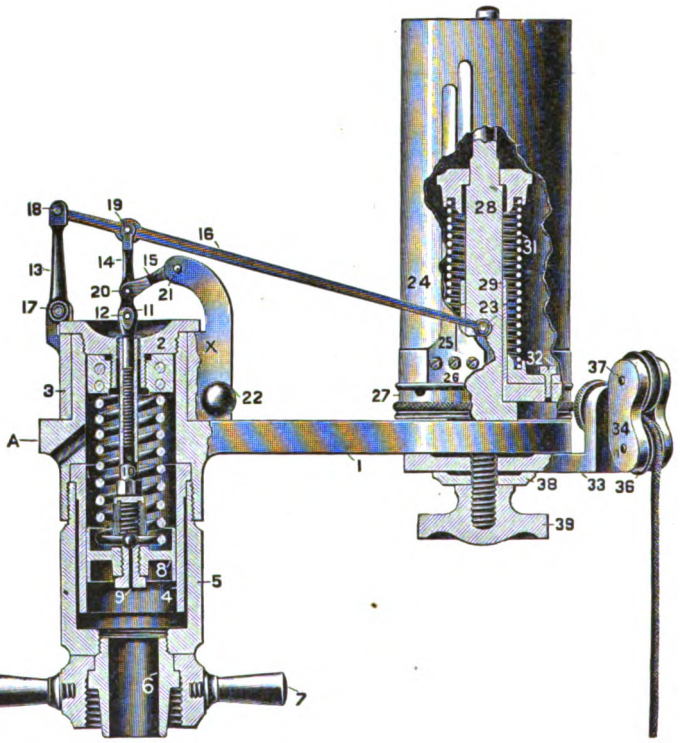

FIG. 10. — The Steam-engine Indicator, Crosby design.

system, with horizontal abscissas showing travel or displacement of the engine piston along its stroke line, vertical ordinates showing the pressure which existed when the piston was at each successive position.

By means of a short pipe, with a special shut-off cock which fits the union coupling at 6 and 7, the indicator is connected to the cylinder of the engine. In Fig. 2 is shown a double pipe connection, with a three-way cock, which enables one indicator to serve both ends of the cylinder; but it is generally better, and as size and speed of engine in-

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crease it becomes decidedly better, to use separate indicators with short and direct pipes. When the indicator cock is opened, steam from the engine acts upon the little piston 8 and compresses the spring above this piston by an amount proportional to the pressure exerted. The pencil mechanism, made up of pieces 13, 14, 15, and 16, magnifies the small piston movement and completes the apparatus for measuring pressure. The spring is so proportioned that it gives to the pressure ordinate a scale of a certain number of pounds per square inch to the inch of rise of the pencil at 23. The diagram is drawn on a slip of paper carried by the paper drum 24. This drum is moved by a cord, wrapped around the pulley 27 and pulling against the spring 31, which is attached to a special mechanism driven by the crosshead and so designed as to give an exact reduced copy of the motion of the engine piston; then as the drum oscillates back and forth upon its axis the paper moves with the piston, and the pencil traces the circuit of the diagram. The length between perpendiculars, projected on the base line MN in Fig. 11,

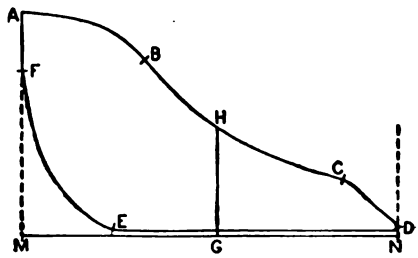

FIG. 11. — The Steam or Indicator Diagram.

represents the stroke of the engine, and this line MN, drawn by the indicator pencil when steam is shut off, is also the line of atmospheric pressure, or the atmosphere line. Ordinates measured from MN, like GH, show the difference between the steam pressure on the under side of the indicator piston and the atmospheric pressure on the top side. When closed to the engine, the indicator cock admits air freely to the cylinder of the indicator.

(*j*) ACTION OF THE STEAM. — In considering the form of the indicator diagram, and the steam action which it shows, we start at the point A and follow the curve in the direction ABCDEF. The piston being at the beginning (the left end) of its stroke, the valve opens the steam port and admits steam to the cylinder, the pressure rising to the height MA. This first part of the "admission" fills the clearance volume, which is made up of the space left between piston and cylinder head (these must not come too close together when the engine is on

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dead center) plus the volume of the steam port. As the piston advances, the valve being well open, there is continued admission of steam, the pressure keeping up close to that in the boiler; but as the valve gradually closes the pressure falls off more or less on account of the choking or throttling of the entering current. This action is shown by the droop of the admission line AB toward the point of cut-off or valve closure at B. With the supply shut off and the piston still advancing, the steam in the cylinder "expands," the pressure decreasing as the volume increases, according to the expansion curve BC.

When the piston is at C the port is opened on the exhaust side and release begins, the steam escaping gradually as shown by the line CD, and dropping almost to the pressure of the atmosphere at the end of the stroke. As the piston returns it expels the low-pressure steam filling the space ahead of it, until the point is reached at E where the valve closes to exhaust; then the steam remaining is compressed into the clearance space, its pressure rising along the compression curve EF. The exhaust line DE, while generally almost or quite a straight line, is always a little above the pressure in the space to which the steam is escaping, just as the admission line AB is always below boiler pressure, because of the resistance to flow offered by the pipes, valve, and ports. In a condensing engine, DE will be well below MN, but yet a little above the pressure in the condenser.

The diagram in Fig. 11 shows the action in one end of the cylinder, or the pressure exerted upon one side of the piston; a corresponding diagram, reversed right and left, would be obtained from the other end.

By finding from the steam diagram the average working pressure upon the piston, we may calculate the work done or the power developed by the engine.

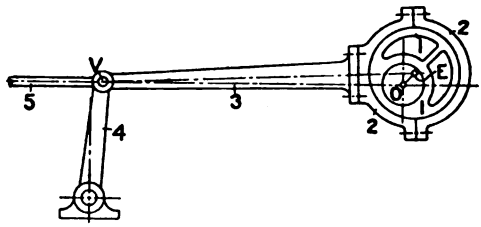

FIG. 12. — The Valve-gear Mechanism.

O. Center of shaft.
1. Eccentric disc or sheave.
2. Eccentric strap.
3. Eccentric rod.
E. Center of eccentric.
4. Rocker arm.
5. Valve rod.

(*k*) THE VALVE GEAR. — In the case of the main engine mechanism, we start with the reciprocating piston and transmit work to the rotating shaft; to drive the valve, this process is reversed, a reciprocating motion

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being derived from the rotary motion of the shaft. The mechanism is essentially the same in both cases, though differing a good deal in the form of its parts.

The driving crank, called the eccentric, has a very short arm or throw; and the crank pin, except where it can be placed off the end of the shaft, as is done in some center-crank engines like Fig. 7, has to be enlarged into an eccentric disc big enough to go around the shaft, as shown in Fig. 12. No matter what the size of this disc, the essential thing so far as motion is concerned is the position of the center E or the length of the radius OE. The eccentric rod is exactly equivalent to the connecting rod; but a rocker arm to guide the joint pin V is far more usual than a slide block, to which it is practically equivalent.

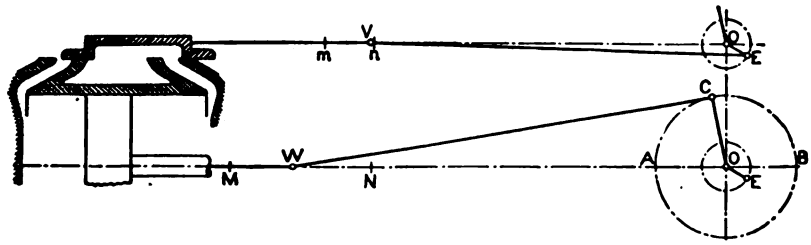

FIG. 13. — Relative Positions of Valve and Piston.

To illustrate the relation between the movements of the two sliding pieces in the engine, the piston and the valve, we combine the outlines of the two mechanisms in Fig. 13, also turning the section of valve and steam passages into this same plane. The simplest form of plain slide valve is used in the drawing. Projected on any vertical plane, the crank arm OC and the eccentric arm OE form a rigid figure COE, which turns about O as a fixed center. For any position of the piston in the cylinder, or of the wrist pin W on its stroke line MN, we measure off WC to locate this crank eccentric, then measure back EV to locate the valve. When the subject of valve action is taken up in detail, short-cut methods for thus determining relative positions will be developed.

(*l*) VALVE ACTION. — This is illustrated by Fig. 14, where the working of the valve is traced out for a revolution of the engine. In diagram I, full lines, the piston is at its extreme left-end position, the crank is on its left or head-end dead center, and the valve is open by a small amount, so that the steam has a chance to enter and fill the clearance space before the piston begins its stroke. As the crank turns in the direction of the arrow — right-hand or clockwise rotation — both piston and valve move toward the right, until the dotted-line position is reached; here the

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valve is at the right-end limit of its movement, and the port has its fullest opening. The valve now returns, gradually diminishing the port opening, until, in the full-line position shown at II, it closes the port or cuts off steam. Expansion takes place while the valve moves toward the left, the dotted position showing where release is just about to begin.

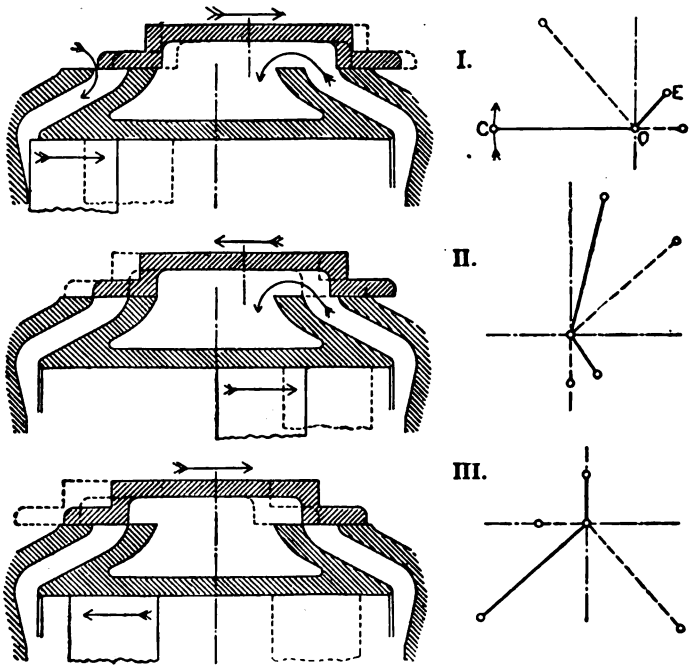

FIG. 14. — Valve Movement.

In III the dotted position shows fullest opening for exhaust; thereafter the valve returns toward the right until, as drawn in full lines, it closes the port to exhaust and starts the compression.

Examination will show that the timing of the events in Fig. 14 does not correspond with that on the diagram in Fig. 11.

(*m*) GOVERNING THE ENGINE. — Any ordinary engine for the generation of power must be provided with some automatic device for controlling its running, so as to keep the speed nearly constant. The smaller and cruder engines have usually a governor which operates a special throttle valve in the supply pipe, and cuts down the working pressure of the steam as less power is needed. In all the higher grades of practice, however, the governor acts to vary the cut-off, making it later or earlier in the stroke as more or less power is called for.

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The high-speed type of engine has a shaft governor, of which a good example is illustrated in Fig. 15. The eccentric is not fastened upon the shaft, but is carried on the swinging piece PQ, pivoted on the wheel at P; movement of the center E with reference to the shaft center O

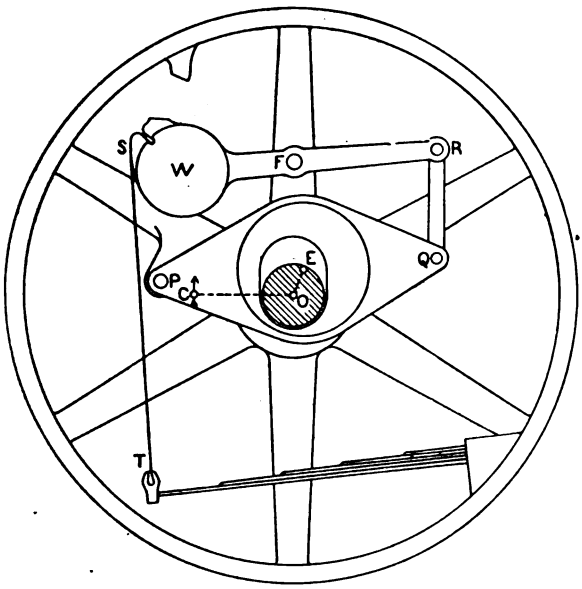

FIG. 15. — The Shaft Governor.

and the crank arm CO produces the desired variation in cut-off (in a manner which is explained in § 39). The controlling device consists of the weighted arm FW and the flat-leaf spring at the bottom of the figure. The centrifugal force of the weight W acts against the elastic force of the spring, transmitted along the steel strap TS. If the load on the engine is increased or diminished, it will slow down or speed up until the change in centrifugal force causes enough movement of the whole governor to accommodate the power of the engine to the new load. The same effect of automatically varying the cut-off is secured by the very different governor gear of the Corliss engine, as described in § 43. In Fig. 3 is shown a second governor, intended to guard against overspeeding. If for any reason the main governor fails to control the engine, an increase of from five to ten per cent above normal speed will cause the safety governor to trip the emergency stop valve, shutting off steam and stopping the engine.

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§ 3. Classification and Characteristics of Engines

(*a*) CLASSIFICATION ACCORDING TO SERVICE. — The influence which most strongly affects the design and construction of an engine, and from which result the most important and essential variations in type, is the kind of service for which the engine is intended. On this basis the following main divisions suggest and justify themselves:

1. *Stationary Engines for the Generation of Power.* — In every case, the engine has a rotary load, or the power is delivered through the shaft. This power may either be transmitted mechanically (by belt or rope) or be first changed into electrical current. For present purposes we place an engine which is direct-connected to an electric generator in this power class rather than in that which follows.

2. *Directly Loaded Stationary Engines.* — The working machine, which applies the power of the engine directly to the useful effect (this does not describe the electric generator), is closely and intimately connected or combined with the engine. Power may be delivered through the shaft, as in a mine hoist or a rolling mill, or through the piston rod, as in pumps and compressors.

3. *The Locomotive.* — With many variations in detail, this conforms closely to one prevailing type.

4. *Marine Engines.* — In modern practice, and for driving screw propellers, these show scarcely any variation in type.

(*b*) OTHER BASES OF CLASSIFICATION. — After service, and of course largely determined by adaptation to its conditions, come general form and arrangement, manner of using the steam, and mechanical features. In the matter of steam working, engines may be simple or multiple-expansion, as already defined in § 2 (*a*); they may use steam of high or low pressure, and either saturated or superheated; and may be run either condensing or noncondensing. As to general form and mechanical features, the engine may be horizontal, vertical, or of special shape; it may have one of several different styles of framework; and may vary quite widely in the number and arrangement of cylinders and cranks. Besides the distinction of simple and compound, there is an analogous comparison of simple and multiplex arrangements; that is, more than one complete engine, whether simple or compound, may be combined in one machine, as in a duplex compound locomotive or pumping engine. The types of valve gear and of controlling or governing apparatus show important differences, and combined with them is the question whether the engine runs in but one direction or is reversible.

To go into the illustration and description of the several types of engines and their variations is beyond the scope of this book. A good

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selection of representative examples will be found in THE STEAM ENGINE, Chapter VIII, Vol. II.

(*c*) THE LAYOUT OF AN ENGINE. — In order to be able to state concisely certain important information as to the arrangement of an engine, we must adopt conventional terms descriptive of position and direction of rotation, as follows:

Right and Left. — The right side and the left side of a horizontal engine are determined by standing back of the cylinder and facing toward the shaft. As to what is meant by right-hand and left-hand, practice is not uniform; but the writer prefers the scheme of calling the side opposite the wheel (in a side-crank engine) the front side, and then going by the right and left position of this front. The engine in Figs. 3, 6, and 8 is thus made "right-hand." In center-crank engines the distinction is less marked and of less importance; it is best simply to specify on which side the governor is placed, and on which side the generator when the engine is direct-connected.

Over and Under. — A horizontal engine runs over if it makes the forward stroke — the piston moving toward the shaft — while the crank traverses the upper part of its circle. If we face a right-hand engine from the right side, it will have clockwise or right-hand rotation when running over.

In a vertical engine, the front side is properly that toward which the crank pin moves when traversing the upper part of its path. Quite often the framework is made heavier at the back, especially in marine engines; but in many other cases it is practically symmetrical.

TABLE 1. SPEED DATA FOR "HIGH-SPEED" ENGINES.

Piston stroke.	Revolutions per minute.	Feet per minute.
Inches.		
12	260 to 300	520 to 600
16	210 to 250	560 to 667
20	180 to 210	600 to 700
24	150 to 180	600 to 720

TABLE 2. DATA FOR ENGINES OF THE CORLISS TYPE.

Piston stroke.	Revolutions per minute.	Feet per minute.
Inches.		
24	85 to 125	340 to 500
30	80 to 115	400 to 575
36	80 to 110	480 to 660
42	75 to 100	525 to 700
48	70 to 90	560 to 720
60	60 to 75	600 to 750

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(*d*) SPEED OF ENGINES. — This is measured in two ways, by the rotative speed or the revolutions per minute, and by the piston speed or the distance in feet traveled by the piston in one minute. The data in Tables 1 and 2 will give a good idea of the usual range in stationary practice.

It is at once apparent that the distinction between the two classes is found chiefly in the rotary speed. Further, the range in piston speed (feet per minute) with any particular stroke is greater in the second table than in the first; this is because the Corliss table shows the variation in practice extending over a much longer period of time than is covered by Table 1.

Accepting for a convenient basis of comparison the usual range from 500 to 750 ft. per min., as set forth in the tables, a wider view of practice in this matter will yield results about as follows:

The lowest piston speeds are found in small steam pumps, such as are used for boiler feeding. With a stroke of 6 in. or less, the proper number of "revolutions" is usually set at 75 per min. as a maximum, making the piston speed about 75 ft. per min. For larger pumps of this type, the limit is fixed at about 100 ft. per min., while the long-stroke pumps without fly-wheel control may rise to 160 ft. per min. Pumping engines with fly wheels come next, usually ranging from 120 to 250 ft. per min.

Air compressors run faster than pumps, though not so fast as power engines. Piston speeds of 300 to 500 are common, while in very quick-running machines as high a figure as 700 ft. per min. may be reached.

Very large power-house engines go above the limits in the tables, speeds of 750 to 900 ft. per min. being common.

In transportation service, where space and weight are considerations of the first importance, the highest speeds are reached. Of course, speeds of 500 to 700 are common in the slower types of both steamers and locomotives; but for fast and full-speed service 900 to 1000 are values often reached and maintained. In very fast locomotives the piston speed often rises as high as 1300 to 1400 ft. per min.

§ 4. The Steam Turbine

(*a*) STEAM ACTION IN ENGINE AND TURBINE. — In the ordinary pressure engine, the elastic force of steam is directly applied to the doing of useful work, through its action upon the surface of the moving piston. The reaction of this working element is just like that of any other part of the confining surfaces within the engine, in that it balances the internal stress in the steam. The steam force is of the nature of a

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static pressure, even though it is exerted upon a surface which moves; and the engine may properly be said to work on the static-force principle.

In the turbine, on the other hand, the expansive force is not exerted upon an external body, but upon the mass of the steam itself, giving to the current or jet a very high velocity. The pressure-work effect is changed into kinetic energy, and the latter is usefully applied in one of two ways: either the jet is directed from a fixed nozzle upon moving curved vanes, in such a way that its resistance to change in direction of flow (by the curved surfaces) will act as a driving force to propel the vanes; or else the jet is formed within the moving element of the machine and the reaction which is opposite to the force accelerating the steam — that is, the recoil of the jet — serves as driving force. In either case, the turbine works on the dynamic-force principle.

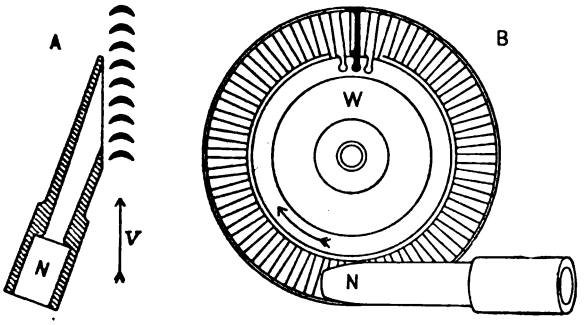

FIG. 16. — Nozzle and Wheel of the De Laval Turbine. View A shows a developed (primarily cylindrical) section through the vanes.

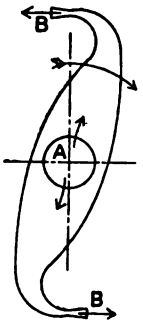

FIG. 17. — Outline of an Elementary Reaction Wheel.

The two working schemes just described are illustrated in Figs. 16 and 17. In the first arrangement — referring especially to view A of Fig. 16 — steam of high pressure enters the outer (lower) end of the nozzle N, and is expanded to exhaust pressure, the wheel spinning in an atmosphere of this low-pressure steam. The action of the jet is obvious, and a turbine in which the vanes thus receive the impulse of a fully formed jet is said to be of the impulse type.

In the reaction "wheel" of Fig. 17, steam of working pressure enters the hollow rotor at A, and jets are formed in the nozzles B, B; these jets blow backward, and their reactions upon the nozzles drive the wheel forward. This jet-driven apparatus is then of the purely reaction type.

(*b*) THE SINGLE-EXPANSION TURBINE. — The simplest steam turbine, thermodynamically, is that in which the steam drops all the way from initial working pressure to exhaust pressure in one operation, or in passing through a single nozzle or set of parallel nozzles. The prominent

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example of this single-expansion or single-stage type is the De Laval turbine, of which the essential form has been outlined in Fig. 16, while Fig. 18 shows the section of a complete machine. The nozzles draw from the steam chamber B, receiving steam of which the working pressure has been fixed by a throttling governor.

With single expansion the steam jet attains a tremendous velocity — something like 2000 to 3000 ft. per sec. — and the vanes must move very rapidly in order to absorb a fair proportion of the kinetic energy of

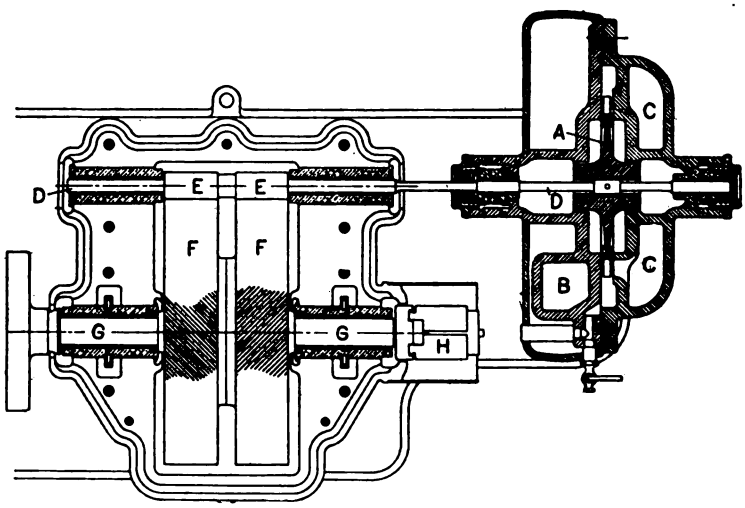

FIG. 18. — Section of a 30-horse-power De Laval Turbine, with wheel about 8 in. in diameter at 20,000 r.p.m.; vane velocity about 700 ft. per sec.

A. Turbine wheel.
B. Steam chamber.
C. Exhaust chamber.
D. Wheel shaft.
E, F. Speed-reducing gears, ratio about 10 to 1.
G. Power shaft.
H. Governor.

this jet. In De Laval turbines comparatively small wheels are used, practice ranging from a 4-inch wheel at 30,000 r.p.m. to a 30-inch wheel at 11,000 r.p.m. To bring these rotary speeds down to something practically applicable, toothed gears are used, with very accurately cut helical teeth, as partly represented on the larger wheel F in Fig. 18. The turbine wheel is mounted on a light, flexible steel shaft, so that it will spin without vibration. Fig. 16 is in correct proportion for a 4-inch, 5 to 7 horse-power wheel, with one nozzle. In the larger sizes a number of nozzles are used, up to twelve in the 30-inch, 300-horse-power machine, the nozzles being disposed about the circumference of the wheel casing.

Single-stage turbines with a very large wheel and no speed-reducing

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gearing have been built and successfully operated; but this type has never been developed beyond the trial stage, because of the superior advantages of the schemes now to be described.

(c) THE MULTIPLE-EXPANSION TURBINE. — The first plan that presents itself for diminishing the required vane velocity is to cut down the velocity of the steam jet. This can be done by dividing the expansion, or pressure drop, or energy transformation, into a number of steps or stages, through the use of a succession of nozzles and vane wheels. The scheme is typified by Fig. 19, where the main view is a development or flattening-out of a cylindrical section through vanes and nozzles.

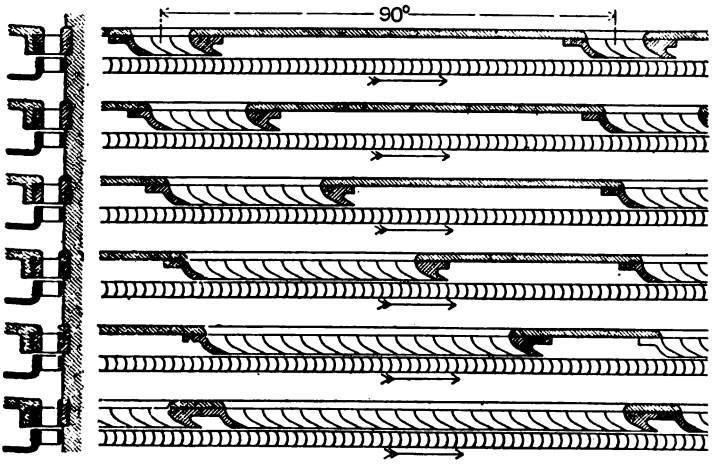

FIG. 19. — Developed Section of Multiple-stage Impulse Turbine with partial peripheral admission; based on high-pressure end of Rateau design, Fig. 20, but with rate of expansion or nozzle increase somewhat exaggerated.

The turbine chamber is divided into a series of cells, in each of which a wheel revolves. The first two wheels at the left end of Fig. 20 are marked R for "rotor"; and at the same place is shown the hollow, box-like construction of the partition discs G, G, which otherwise are cross-hatched as if solid, in order to emphasize the difference between their bulk and the open space which is filled with steam. There is, of course, a running "joint" between each fixed disc and the turbine shaft, which is made as nearly steam-tight as possible.

In Fig. 20, steam enters from the governor valve at A, passes by B to C, goes through the first group of wheels to D, and through the second group to E; thence a connecting pipe carries it to the low-pressure "cylinder" at F, and finally it flows from H to the condenser. The valve B is an emergency device, to enable the turbine to meet an overload; when it is opened, there is a larger passage for high-pressure steam

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than is afforded by the small nozzle area at C, with a consequent increase in steam admitted and power developed, but with some falling off in efficiency of operation.

This example, chosen as an extreme case of subdivision into stages, is an early design, and has a larger number of wheels than fuller experience has shown to be desirable; and, except in marine service, the idea of dividing the turbine into separate sections has generally been given up. As in other lines of machinery, the tendency in the development of the turbine is toward less complex forms.

Note how provision is made for increasing the cross area of the steam channel as the pressure falls and a current with a certain velocity needs

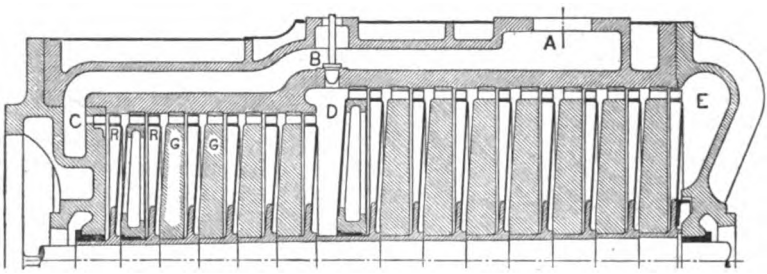

FIG. 20. — Section of a 24-stage Rateau Turbine, in three steps and two cylinders; 500 horse-power at 2400 r.p.m.; mean diameter of vane rings, 20 in. to 33 in.; vane velocity, from 220 to 345 ft. per sec.

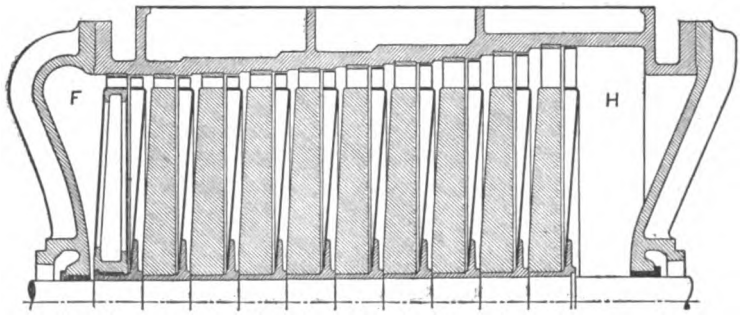

FIG. 20. — Continued. Low-pressure section or cylinder of the turbine, on same shaft with high-pressure section.

more room, because of the expansion of the steam. In Fig. 19 the scheme of partial peripheral admission, with increase in width of nozzle opening, is shown. The second part of Fig. 20 shows how, after the whole circumference has been taken up, the radial dimension of nozzles and vanes is progressively increased.

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(*d*) THE MULTIPLE-IMPULSE TURBINE. — In logical sequence, after the idea of dividing the operation of jet formation or kinetic-energy development into a number of stages comes the scheme of similarly dividing the energy absorption or jet application. A typical embodiment of this idea is shown in Fig. 21. The jet is fully formed in the

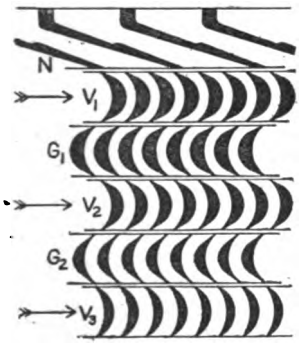

FIG. 21. — Three-impulse Element of a Curtis Turbine.

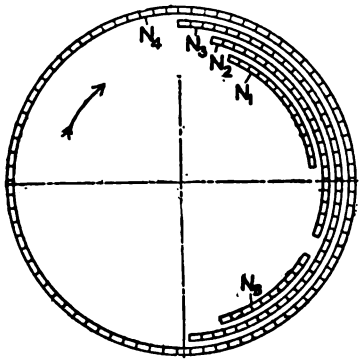

FIG. 22. — Diagram of the Nozzle Admissions in the Curtis Turbine.

nozzles N; passing through the first vane row V_1, it is discharged into the fixed guide vanes G_1, which deflect it back into the proper direction for driving, and deliver it upon the second row V_2 of moving vanes; and after a repetition of this operation the steam is finally discharged from the vanes V_3. In each set of moving vanes a part of the kinetic energy is abstracted from the jet, its velocity being reduced after a manner which will be fully explained in Chapter IX. To the parts of this process the name "velocity stages" is frequently given, as distinguished from the pressure stages in expansion; but these terms are rather awkward for general use, and it seems better to let "stage" stand for a pressure-drop division, and to use "impulse" for the action of a jet upon any single row of moving vanes.

The Curtis turbine is the most prominent member of the multiple-impulse class. An example is sectioned in Fig. 23, where the general form is clearly shown, with much of the larger detail. Each of the wheels carries two rows of vanes, instead of the three in Fig. 21. The wheel cells are separated by cast-iron diaphragms. The steam supply is controlled by valves at the inlet A, while at C is an automatic by-pass valve, somewhat similar in function to that at B in Fig. 20. A mechanical detail of great importance is the large foot-step bearing at F and G, which carries the combined weight of the turbine and generator rotors, the electric generator being right above the steam turbine.

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In Fig. 22 is given a diagram showing the application of the principle of partial peripheral admission in this turbine, the nozzle openings being represented by the blocked arcs, which are made of decreasing diameter in this sketch merely to keep them from overlapping. The initial nozzles N_1 subtend an arc of perhaps 60°, while the last set N_4 (in the third diaphragm) covers the whole circumference. The short set N_B in the first diaphragm is served by the by-pass valve at C in Fig. 23.

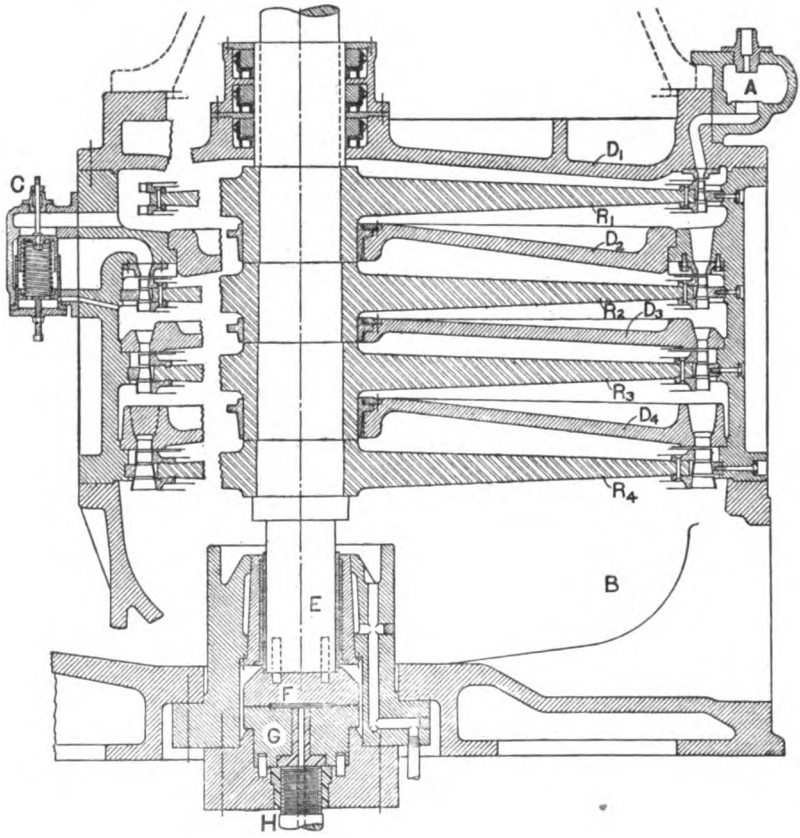

FIG. 23. — Section of a 2000-kilowatt Curtis Turbine, with four two-impulse stages; 100-inch wheels at 750 r.p.m.; vane speed about 325 ft. per sec.

(*e*) THE REACTION TURBINE. — The scheme outlined in Fig. 17 has not been developed into a practical steam turbine, because of certain inherent disadvantages: it would be rather difficult to introduce steam of full pressure into the rotor without undue leakage and friction at the

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running joint, and the embodiment of pressure staging would be very hard to effect.

The reaction principle finds an extensive application, however, in the Parsons turbine, which has very largely prëempted the possibilities along this line of development. The general arrangement of a typical

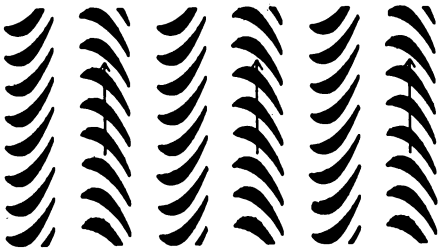

FIG. 24. — Element of the Parsons Turbine.

Parsons turbine is shown in Fig. 25, while Fig. 24 gives the shape of the vanes or blades. The two most essential features are: first, the mounting of the blades in a succession of similar rows, alternately on drum-shaped rotor and on inside of casing, so as to form a continuous passage for the steam, along which its pressure drops progressively and gradually; and second, the shape of the vanes, which depart radically from the symmetrical profile shown in Figs. 16, 19, and 21. In Fig. 24 the course of the steam is from left to right, in a general sense across the vane rows. Each set of vanes is formed to receive a current coming squarely in sidewise, but to deliver this current, after acceleration, in a direction swung well around toward the line of vane movement. Pressure drop and acceleration take place in each vane row, whether fixed or moving. Speaking approximately, the fixed vanes deliver a current at a velocity equal to that of the moving vanes, so that the steam can pass right into the channel entrances between the latter; and the moving vanes deliver backward steam which leaves the channels with a relative velocity about equal (but opposite) to the speed of the vanes themselves, or with an absolute velocity of nearly zero. We here refer to component velocity in the direction of vane movement, as distinct from the general progressive flow across the vane rows, or parallel to the axis of the rotor. The whole matter is discussed and graphically exhibited in Chapter IX, where it is shown that this turbine is not purely of the reaction type, but that there is also some little impulse exerted by the current in entering the moving vane rows.

(*f*) THE PARSONS TURBINE. — Considering now the general drawing in Fig. 25, we note one important characteristic in the enlargement of

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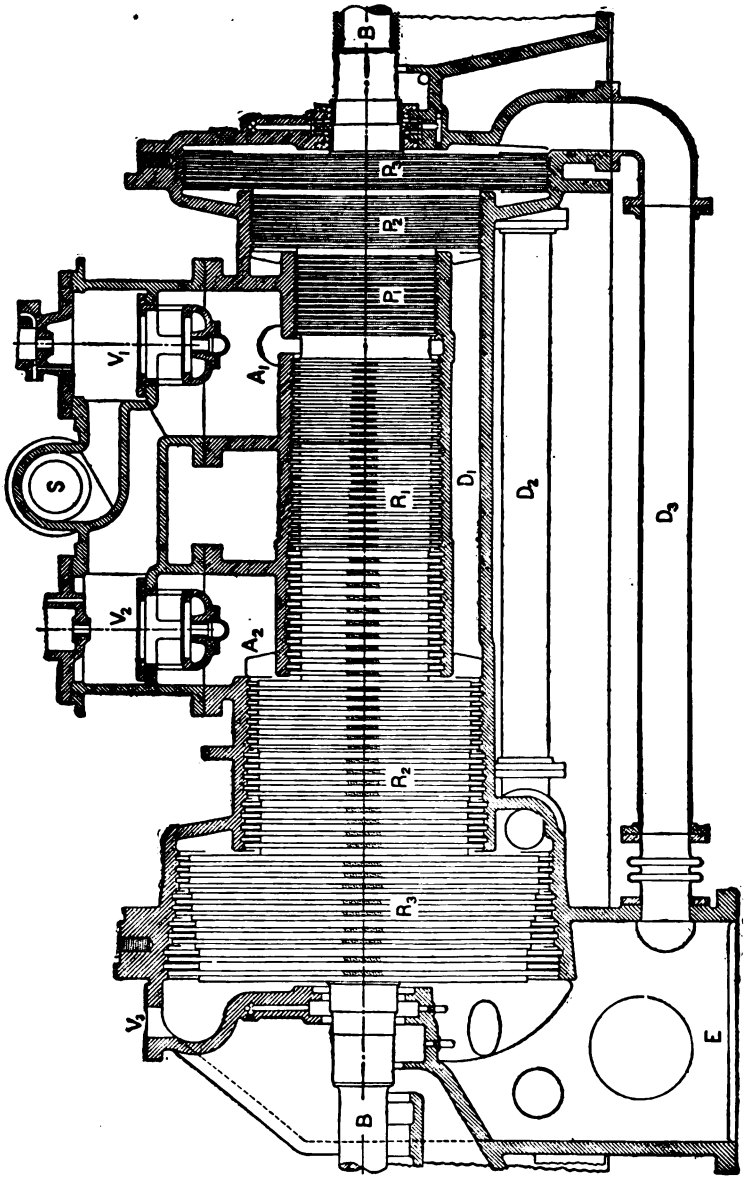

FIG. 25. — Section of a Typical Westinghouse-Parsons Turbine, with 59 stages in three steps.

the rotor toward the low-pressure end: here the steam flows from right to left. This enlargement, together with the increase in vane length, provides the necessary increase in cross area of steam channel. On the annular side of each step or section there will be a steam pressure which

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will tend to force the whole rotor toward the left; further, there will be a higher pressure on the right or entrance side of each vane row than on the left or discharge side, and thus an additional force toward the left. To neutralize these forces, the balance discs or "pistons," P_1, P_2, P_3, are put on the right end of the rotor, and pressures are equalized through the passages D_1, D_2, D_3. Since but half of the total pressure drop takes place in the rotor vanes, the balance-piston diameters extend only to mid-length of the vanes. Leakage is minimized by a collar-and-groove surface of piston and casing, forming what is called a labyrinth packing (see Chapter X).

The main controlling valve (a double-seated valve of the form II in Fig. 280) is at V_1, and normal steam admission at A_1. Under overload the by-pass valve V_2 opens, and steam enters directly the larger through-vane channel at A_2; this throws the first step R_1 pretty thoroughly out of action, its vanes merely churning steam, and there is some consequent loss of efficiency.

(*g*) Turbines of Mixed Type. — In a Parsons turbine there must be a slight clearance between the ends of the blades on one part (rotor

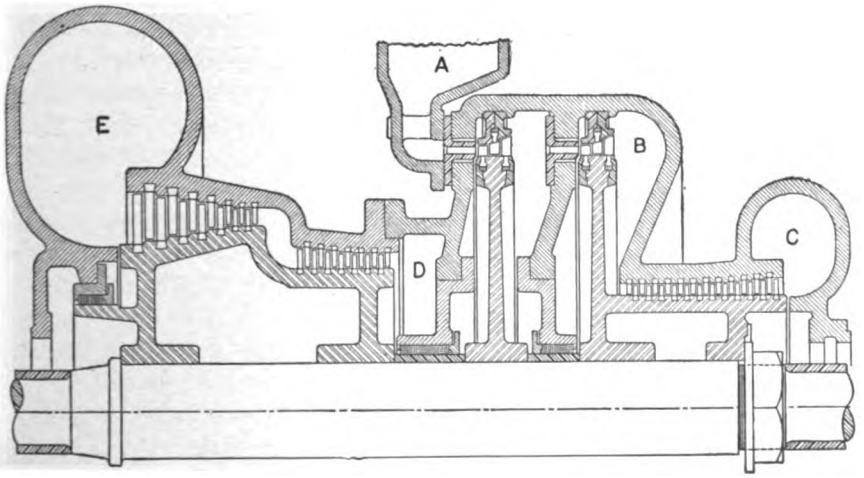

Fig. 26. — Half Section of the Sulzer Turbine.

or casing) and the surface of the other part, for it would not do to have actual contact and rubbing. This leaves a passage for leakage from stage to stage, which may be of considerable relative amount at the high-pressure end of the machine. For this and other reasons, a number of composite designs have been developed, of which Fig. 26 is an excellent example. There are, first, between A and B, two wheels of the Curtis

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type, with two-impulse arrangement; then, from B to C and from D to E, there are three groups of Parsons-type blades. These are so placed that their end thrusts equalize each other, and no balance-piston device is needed.

The foregoing description is intended to give a general idea of the form and working of the principal types of turbines. To sum up, the characteristic arrangements are as follows:

A. One pressure stage and one velocity stage, or the single-stage single-impulse turbine.

B. A number of pressure stages, each with one velocity stage, or the many-stage single-impulse turbine.

C. Several pressure stages, each with several velocity stages, or the few-stage multiple-impulse type. Small turbines of this class have sometimes but one pressure stage; the common range, however, is from two to five stages.

D. The many-stage reaction type, necessarily without velocity staging.

E. Combinations of type C with B or of C with D.

The illustrations here, and those in Chapter X — which will show some further variations in general form and arrangement, as also the details of construction — are from turbines of the stationary type, used for driving electric generators. In marine service there are the two special requirements of comparatively low speed and of reversibility. The first is met by increasing the number of stages, whether in pressure or in velocity; while for the second an auxiliary section is added to the rotor. This backward turbine, of a few stages and hence of low-working efficiency, is at the low-pressure end of the main turbine, and normally runs idle in an atmosphere of steam of the condenser pressure.

A recent development is the use, on a large scale, of gearing similar to that of the De Laval machine. This is being worked out for ship propulsion, and has also been tried, with promise of success, for the driving of machinery. The gears must be very accurately cut, and the difficult requirements are those of quiet running and of durability.

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CHAPTER II

ELEMENTARY THEORY OF THE HEAT ENGINE

§ 5. Heat and Work

(*a*) THERMODYNAMICS is the science of the relations between thermal and mechanical energy, or between heat and work, and is especially concerned with the transformation of one of these forms of energy into the other. This transformation can be made in either direction, and always takes place in a fixed and definite quantitative ratio; but while the whole of a given supply of mechanical work can easily be changed into heat, it is inherently impossible to convert into work more than a certain proportion of a given supply of heat. An apparatus for turning heat into work is called a heat engine; and the final object of this chapter will be to show the manner of operation and to find the limiting performance of the best possible heat engine.

In every such apparatus, heat conversion is effected through the agency of an expansive fluid or working medium, which alternately expands and contracts as heat is given to and taken from it. Typical substances are, air (as representing the dry-gas mixture in engines of the internal-combustion class), and steam (or the liquid-to-vapor medium). The former follows much the simpler physical laws in its behavior, and we will therefore base upon it our primary development of thermodynamic ideas and principles.

(*b*) HEAT is one variety of energy. As existent in sensible* form, in material bodies — the form in which it is directly measureable — it is believed to consist in a vibratory motion of the molecules of the body, and is therefore a sort of kinetic energy. Heat may also be stored in a substance in a latent or potential form, as when it changes the substance from liquid to vapor. Radiant heat, or the wave action of the ether by virtue of which heat traverses space, is of no direct interest in thermodynamics. It is assumed that the reader has a working knowl-

* Sensible heat is that which shows its presence by temperature or its addition by raising the temperature of a body: in other words, it is heat that can be felt.

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edge of the physics of heat, and knows what is meant by radiation, conduction, temperature, specific heat, etc.; but we now wish to lay renewed emphasis upon the two ideas of intensity and quantity of heat.

Heat intensity is only another name for temperature: the higher the temperature of a body, the more rapidly do its molecules move, and the more energy do they possess because of this motion. The intensity or temperature of a supply of heat is a controlling determinant of its availability for transformation into work, as will appear in the development of this discussion. The unit of temperature which we shall use is the fahrenheit degree.

The quantity of sensible heat in a given body, above any chosen reference state, depends upon the temperature of the body and its heat capacity, the latter being the product of the weight of substance and the specific heat or unit capacity. Our practical unit of quantity is the British thermal unit or B.t.u., which we shall define as the $\frac{1}{180}$ part of the heat required to raise one pound of pure water from 32 deg. to 212 deg. fahr. The pound-degree capacity of any substance (including water at various temperatures), measured in terms of this unit, is identical with its specific heat.

The temperature-by-capacity measure of heat is not always directly applicable, for under certain conditions large amounts of heat can be put into a substance without affecting its temperature. For the convenient representation of thermodynamic operations, it has been found necessary to invent another second factor to go with temperature. This is called entropy, and its form and use will be brought out presently, in § 9.

(*c*) WORK AND POWER. — For mechanical purposes, work is to be defined as the overcoming of resistance through distance; or, to make the action subjective, when a force acts upon a moving body, in the direction of the motion and against an equal and opposite resistance, it does work upon that body of which the amount is equal to the product of force by distance. The common unit of work is the foot pound, the work done in overcoming one pound of gravity resistance through one foot, or its equivalent.

Measurement of the work performance of a machine is usually expressed, not in terms of absolute work quantity, but rather by the work rate, or what is called the "power" of the machine. This brings in the element of time, so that if work is defined as force by distance, power becomes force by velocity. For small measurements, the foot pound per minute or the foot pound per second may be used; for the engine, the horse-power or h.p. is the practical unit. This is an arbitrarily established rate of 550 ft. lb. per second, or 33,000 ft. lb. per minute,

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or 1,980,000 ft. lb. per hour — these numbers all embodying the same ratio of work to time.

For large amounts of work, especially in comparing the work done by an engine with heat received, convenient quantity units are derived from the horse-power rate, by isolating the output in a unit of time. Thus one horse-power in one minute does 33,000 ft. lb. of work, called the horse-power-minute or h.p.m.; similarly, a horse-power-hour or h.p.h. is an absolute quantity of 1,980,000 ft. lb.

(*d*) Relation Between Heat and Work. — The mechanical equivalent of heat, or the ratio at which heat and mechanical energy are interchanged, is shown by the equation,

$$1 \text{ B.t.u.} = 778 \text{ ft. lb.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

For the number 778 the symbol J is generally used, so that if we have a quantity Q of heat energy, its value U in foot pounds will be

$$U = 778\, Q = JQ. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Conversely, to express work in heat units, we use the letter A for the heat equivalent of work, so that

$$Q = \frac{1}{778} U = 0.001285\, U = AU. \quad . \quad . \quad . \quad . \quad (3)$$

In calculating the efficiency of heat engines, or finding the ratio of work done to heat supplied, it is usually most convenient to reduce the work output to heat measure; and for this purpose it is well to remember that one horse-power-minute is equal to 42.42 B.t.u., and one horse-power-hour equal to 2545 B.t.u.

(*e*) Thermodynamic Operations. — In every thermodynamic operation with an expansive medium, there are two sides or view-points from which it may be regarded. One is the mechanical side, with pressure and volume as the prominent quantities; the other is the thermal side, with emphasis on temperature and heat quantity. In § 7 (*d*) it is shown that pressure and volume can be used as factors in work quantity, in place of force and distance. In § 9 the idea of entropy is developed, with the resulting method of graphical representation of thermal quantity. At first we shall approach the subject rather from the side of mechanics.

§ 6. The Perfect Gas.

(*a*) The Perfect Gas. — The so-called permanent gases, such as hydrogen, oxygen, nitrogen, and air, which under ordinary conditions are very far from their state of liquefaction, follow very simple laws when heated, cooled, expanded, or compressed. Under precise observation, however, even these gases show small irregularities and departures

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from the simple laws. As a basis of fundamental theory, we assume, therefore, an ideal, perfect gas. This is supposed to be made up of homogeneous and physically indivisible molecules, with no attractive forces acting among them, and separated by distances which are exceedingly large in comparison with their own dimensions. These molecules are in constant, free, rapid, rectilinear motion, rebounding with perfect elasticity as they impinge upon each other or upon the walls of a confining vessel.

(*b*) LAWS OF THE PERFECT GAS. — Discovered originally through experiment, but later derived mathematically from the physical definition just given, the general law of behavior of the perfect gas is expressed by the equation

$$\frac{pv}{T} = C. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Here p = pressure, measured above the zero of perfect vacuum;
v = volume of a given, definite quantity (weight) of the gas;
T = temperature, measured above what is called the absolute zero, which, on the fahrenheit scale, is 460 degrees below the ordinary thermometer zero;
C = a constant.

From this general, three-variable equation we get simpler relations by holding constant, one at a time, the three quantities p, v, and T. Thus if p is kept constant,

$$\frac{v}{T} = C \quad \text{or} \quad v = CT, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

or, volume and temperature vary together at a constant relative rate.

Similarly, if volume v is kept constant,

$$\frac{p}{T} = C \quad \text{or} \quad p = CT, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

or, volume and temperature vary together at a constant relative rate.

Finally, if temperature T is kept constant,

$$pv = C \quad \text{or} \quad p = \frac{C}{v}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

or, pressure varies inversely as volume.

The first two relations, Eqs. (5) and (6), are the two forms of the law of Gay Lussac; the third, Eq. (7), is the law of Boyle or of Mariotte. It must be understood that the symbol C is here used in the most general way, standing for a different value in each of the four equations that have been given.

(*c*) THE LAW OF GAY LUSSAC. — Change of volume with temperature under constant pressure, Eq. (5), or of pressure with temperature at constant volume, Eq. (6), involves, of course, the addition or abstraction of heat; and the two operations are generally known as heating (or

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cooling) at constant pressure and at constant volume. At this point, however, we consider only the relations between the determining variables p, v, and T, deferring all questions as to quantity of heat or work. The mechanical side of both operations is represented graphically in Fig. 27, where volumes are measured horizontally as abscissas, pressures vertically as ordinates. Change at a constant pressure p_1 from an initial volume v_1 to a final volume v_2 is shown by the line AB; similarly, increase of pressure from p_1 to p_2 at the constant volume v_1 is shown by

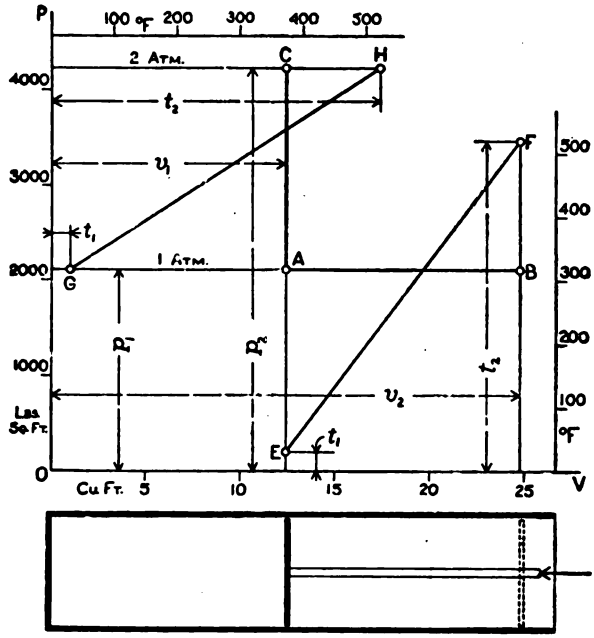

FIG. 27. — Simple Heating Operations.

AC. If the body of gas were confined in the cylinder outlined at the bottom of the figure, the piston would be allowed to move out against a constant resistance in the first case, but would be held fast in the second.

The line EF, on the volume base OV and with ordinates to the scale at the right, shows how temperature rises as volume increases: line GH, on the pressure base OP and to the scale at the top of the figure, is a similar diagram for the case of heating at constant volume. The relation between volume and temperature in one case, and between pressure and temperature in the other, is the same, and it will be necessary to discuss but one condition. In Fig. 28, temperature is the base, volume the ordinate. The part of this diagram from OV toward the right represents experimental data: if the volume of a body of gas at freezing

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point to be taken as unity, the volume at boiling point, after heating under constant pressure, will be 1.366; and the law of variation is shown by the straight line AC, corresponding with Eq. (5).

(*d*) THE ABSOLUTE ZERO. — If now, in Fig. 28, the line AC is extended toward the left, it will cut the base line TO at a point Z, which is called the absolute zero; and from this point absolute temperature

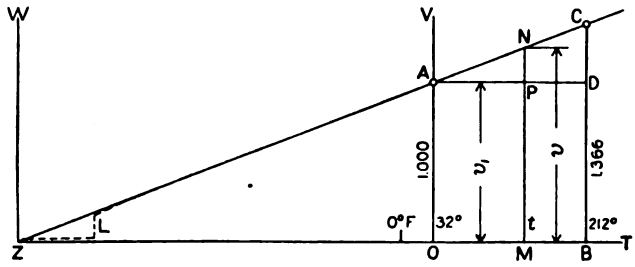

FIG. 28.

is measured. From the relations geometrically set forth in the figure we can get a very good concept of the meaning of absolute temperature, according to the following line of reasoning:

Let the temperature of melting ice, 32 deg. fahr., be taken as a natural and obvious starting point for temperature measurement. Then the triangle ADC or APN shows the ratio, definite and constant, between volume increase and temperature rise. Consider now the volume v at any temperature t on the thermometer scale: by making the triangle ZMN similar to APN, we apply this same ratio to the whole volume. It is much simpler to have this whole volume MN or v thus proportional to the absolute temperature ZM or T than to have it made up of an initial volume MP or v_1 and an increment PN proportional to the temperature rise AP.

According to Eq. (5) or to the line ZC on Fig. 28, the volume of the gas would be nothing at absolute zero. Of course, no actual gas would thus shrink to nothing. Rather, it would depart more and more from the "perfect" state, and behave as indicated by the dotted profile near Z, where the vertical portion L shows liquefaction, followed by a very slow contraction with further abstraction of heat. The important fact is, however, that over ordinary ranges of temperature the nearly perfect gases follow the straight line law AC; and by measuring temperature from Z we get much simpler relations than if any other reference point were used.

(*e*) ABSOLUTE TEMPERATURE. — Taking from Fig. 28 the proportion

$$ZO : OA :: AD : DC,$$

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letting T_0 stand for ZO or the absolute temperature of freezing point, and substituting the numerical values marked on the figure, we have

$$T_0 : 180 :: 1.000 : 0.366,$$

whence $T_0 = 492$ deg. This locates the absolute zero at 460 deg. below the fahrenheit zero, as stated in the definition under Eq. (4); and between absolute and thermometer temperature we have the relation,

$$T = t + 460 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

(*f*) THE COEFFICIENT OF EXPANSION. — The ratio between the volume change per degree and the total volume of the gas is called the coefficient of expansion. In general, if we have a volume v at a temperature T, the gas changes under constant pressure as though it would shrink to zero in cooling through T degrees; therefore the ratio* of change, which we shall call a, is

$$a = \frac{1}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

In terms of the volume at any temperature T, this coefficient a is a variable, since it is the ratio of the constant increment to the variable total volume. To make it a useful constant, we must choose some particular volume as a base. Usually, the volume at 32 deg. fahr. is taken as unity; then $a = 1 \div 492 = 0.002033$; and the same result is got by dividing 0.366 of volume change by 180 deg. — these numbers coming from the triangle ADC on Fig. 28. Using symbols as marked on that figure, we get very readily the formula

$$v = v_1 [1 + a\,(t - 32)]: \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

but in most cases it is more convenient to use the relation, from Eq. (5),

$$\frac{v_2}{T_2} = \frac{v_1}{T_1} \text{ or } \frac{v_2}{v_1} = \frac{T_2}{T_1}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

where the subscripts are used in a general way, to indicate any two particular conditions.

As remarked in the last paragraph of Art. (*c*), there is no occasion for separately developing formulas to cover the operation of heating at constant volume; a simple substitution of p for v in Eq. (10) or (11) gives all that is needed.

(*g*) MARIOTTE'S LAW. — Expansion (or compression) at constant temperature, Eq. (7), is represented graphically by the curve called the equilateral hyperbola, drawn in Fig. 29. If the operation is to be carried out in a cylinder with piston, as indicated under Fig. 27, the force along the piston rod must vary, always just balancing the pressure of the gas.

* That is, the ratio of the decrement per degree to the initial volume.

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Since this curve must frequently be plotted, the most convenient construction for finding points upon it is given in Fig. 29. Having the reference axes OV and OP (the lines, respectively, of zero pressure and zero volume), and a point A through which the curve is to pass, we draw through A the lines AD and AE, parallel to OV and OP; then drawing

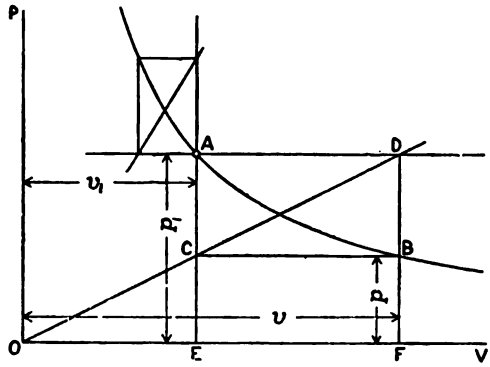

FIG. 29.—The Equilateral Hyperbola.

from the origin O any radial line to cut these auxiliary axes at C and D, and completing the rectangle ADBC, we determine the point B of the curve. For, in the similar triangles DOF, COE,

$$\text{DF or AE} : \text{CE or BF} :: \text{OF} : \text{OE},$$

or

$$p_1 : p :: v : v_1,$$

which makes $pv = p_1v_1$ and satisfies Eq. (7).

An operation which takes place at constant temperature is called an isothermal operation; and the curve in Fig. 29 is the isothermal curve of a perfect gas.

(*h*) FORMULAS FOR AIR. — In order to establish the value of the constant C in the equation $pv = CT$, we must know a full set of coincident values of p, v, and T. For the "standard" conditions

p_0 = atmospheric pressure, or 14.7 lb. per sq. in., and
T_0 = freezing point, or 492 deg. fahr.,

it has been found by experiment that the volume of one pound of air is

$$v_0 = 12.385 \text{ cu. ft.}$$

Substituting in Eq. (4) we get,

$$C = \frac{14.7 \times 12.385}{492} = 0.37004;$$

whence

$$pv = 0.37\,T. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

The value of C depends, of course, upon the units of measurement used, but it is always calculated for the unit of weight, as here the pound.

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The volume of one pound, v in Eq. (12), is called the specific volume of the gas.

Suppose now, as an example, that we wish to see how this specific volume is related to the temperature, at the standard atmospheric pressure; substituting in Eq. (12) the particular value $p = 14.7$, we get

$$v = 0.02517T, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

as the desired formula.

PROBLEMS

1. Find the volume of one pound of air under a pressure of 40 lb. per sq. in. and at a temperature of 80 deg. fahr.

2. A receiver of 40 cu. ft. capacity is filled with air at 60 deg. fahr. and a pressure of 20 lb. per sq. in.; what proportion of it must be allowed to escape in order that the pressure may remain unchanged when the air is heated to 200 deg.?

3. Find the coefficient of expansion under constant pressure if the volume at 212 deg. fahr. is taken as base.

4. For a confined body of air, $p = 40$ lb. and $t = 60$ deg.; what will p be if the temperature is raised to 350 deg., volume being constant?

5. If a receiver of 32 cu. ft. capacity is filled with air at 80 lb. per sq. in. pressure and at 96 deg. fahr., what weight of air does it contain?

§ 7. Simple Thermodynamic Operations with Gases

(*a*) Effects of Heat. — When heat is imparted to an expansive substance, with accompanying changes in pressure and volume, there are three ways in which the heat may be used or applied; these are,

First, in changing the temperature of the body: that is, the heat is directly stored in the body, in the form of increased thermal or vibratory energy of the molecules, without any departure from the sensible condition.

Second, in doing internal work within the substance, by overcoming molecular attractions. Since such work effects changes in the relative positions or arrangements of the molecules, it is called disgregation work. In a perfect gas, with no molecular attractions, this element would be absent; but when there is a change of physical state, as from liquid to vapor, disgregation work is of predominant magnitude. Heat used up in this manner changes from the state of active energy to that of passive or potential energy; it ceases to be sensible heat, but is ready to reappear in that form whenever the conditions are suitable for its escape.

Third, in doing external work of expansion, by overcoming the resistance of the confining surface (of whatever form) through a certain distance. The heat thus used definitely ceases to exist as heat, being transformed into mechanical work or energy; but it is capable of reconversion by a reversal of process.

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Let I stand for internal work, or change of internal energy, either in sensible heat or as disgregation work, or both, and U for external work in foot pounds, so that AU is the same quantity reduced to heat units; then if Q is a general symbol for heat imparted, we have the fundamental equation

$$Q = I + AU. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

We will now consider the thermal relations and quantities involved in several of the simplest and most important operations with gases, using air as an example and giving numerical values.

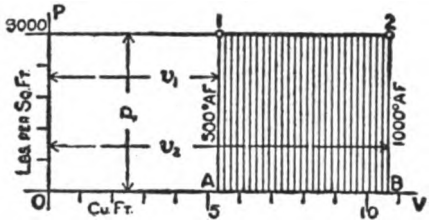

FIG. 30. — Expansion at Constant Pressure.

(*b*) EXPANSION AT CONSTANT PRESSURE. — In Fig. 30 is represented the operation of expanding one pound of air from volume v_1 at temperature t_1 to v_2 at t_2, under the pressure p, the external work being shown by the shaded rectangle 12BA. The amount of this work can be most simply calculated as the product of pressure in pounds per square foot by volume change in cubic feet — see Art. (*d*), following. The pressure p being measured in pounds per square inch, this gives

$$U = 144p\ (v_2 - v_1) \text{ ft. lb.;} \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

and substituting from Eq. (12) we have

$$U = 144 \times 0.37\ (T_2 - T_1) = 53.3\ (T_2 - T_1). \quad . \quad . \quad . \quad (16)$$

That is, for each degree that the temperature is raised, 53.3 ft. lb. of external work is done by the gas, equivalent to $53.3 \div 778 = 0.0685$ B.t.u. This latter number may well be called, by analogy, the specific heat for external work in the particular operation of heating under constant pressure.

In this process, it has been found by experiment that the total amount of heat that must be imparted in order to raise one pound of air one degree is 0.2375 B.t.u. Using for this specific heat under constant pressure the symbol c_p, we have

$$Q = c_p\ (t_2 - t_1) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

as the heat supplied in the operation of Fig. 30.

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Subtracting from the total specific heat c_p the amount expended in external work, we get the rate of heat absorption in change of internal energy in the gas to be,

$$c_v = c_p - 0.0685 = 0.2375 - 0.0685 = 0.1690. \quad . \quad . \quad . \quad (18)$$

(*c*) THE SPECIFIC HEAT FOR INTERNAL WORK, called c_v in Eq. (18), is also the specific heat at constant volume, because under the latter condition the heat supplied does nothing but the internal work of raising the temperature of the gas. A most important fact about this internal work is that it depends wholly upon the initial and final temperatures, and not at all upon the character of the process through which the gas passes. No matter what else happens to a perfect gas in a thermodynamic operation, the change in internal energy and the heat required to produce it is

$$I = c_v (t_2 - t_1). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

Note that c_v is a general constant, while c_p and 0.0685 are special constants, belonging to the operation of heating under constant pressure.

(*d*) THE PRESSURE-VOLUME MEASURE OF WORK. — The method used in Eq. (15) for calculating mechanical work is based upon the following considerations:

If we imagine the body of gas to be enclosed in a cylinder with an area of cross section (or of piston) of one square foot, then every cubic foot of volume change will cause the piston to move one foot, and this movement will be against the pressure per square foot, which is $144p$ or P; evidently, then, $P(v_2 - v_1)$ meets literally the definition of work as force by distance. As the general case, suppose a piston of area A to move the distance S under the pressure P; the work done will be, when expressed as force by distance,

$$U = PA \times S;$$

and we need only change the grouping of the factors in order to get

$$U = P \times AS = P(v_2 - v_1), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

or, work = pressure × volume displaced.

Note particularly that in this pressure-volume method of computing work, all measurements must be in terms of the factors of the work unit, or must be reduced to those terms. Thus in the foot-pound system, pressures must be in pounds and on the square foot, volumes must be in cubic feet. For pressure per square foot we shall use the symbol P, in all thermodynamic formulas; and for the convenient use of Eq. (12) in this connection — as for substitution in Eq. (15) — we give it the form

$$Pv = 53.3\, T = RT. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

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Further, between the two specific heats of air we have the relation

$$c_v = c_p - AR, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

this AR representing for air the number 0.0685, used in Eq. (18).

(*e*) THE CALCULATION OF EXTERNAL WORK. — In the discussion of the operation of heating under constant pressure, we derived the internal work I from experimentally determined values of Q and AU. Having thus established the general expression for I, we follow in other cases the method of calculating the external work AU from data as to pressure and volume, then adding AU to I in order to get Q. Since pressure now varies, the external work must be found by calculating or measuring the area under a curve. An example is represented in Fig. 31; and the element of work performed during the very small displacement dv, which may be considered as taking place at a constant pressure P, is shown graphically to be Pdv. The curves used in theoretical discussions permit of mathematical integration; those from actual engines, drawn by the indicator, must be integrated mechanically, as by the planimeter or by averaging ordinates. Usually, the mean pressure throughout a certain volume change is found, whereupon

$$U = P_m (v_2 - v_1); \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

but sometimes a work scale per unit of area is used, like that shown in Fig. 31, where the unit of base represents 1 cu. ft. and the same unit of ordinate 1000 lb. per sq. ft., so that the square unit is 1000 ft. lb.

(*f*) ISOTHERMAL EXPANSION. — As stated in § 6 (*g*), this operation takes place, with a perfect gas, under the law $Pv = C$. Having the

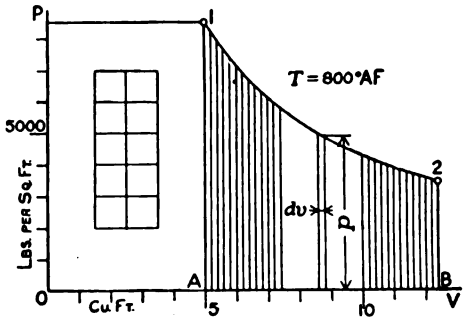

FIG. 31. — The Isothermal Curve.

relation $Pv = P_1v_1$, and desiring to integrate Pdv between the limits v_1 and v_2 which locate the ordinates A1 and B2 on Fig. 31, we get P in terms of v as

$$P = \frac{P_1v_1}{v},$$

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whence

$$\int_{v_1}^{v_2} P dv = P_1 v_1 \int_{v_1}^{v_2} \frac{dv}{v} = P_1 v_1 \log_e \frac{v_2}{v_1}$$

or

$$U = Pv \log_e r. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

Here r is the ratio of expansion, always greater than unity (and therefore the inverse ratio of compression), and is equal to $v_2 \div v_1$ or $P_1 \div P_2$. Substituting from Eq. (21), we have

$$U = RT \log_e r. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

Since there is no change in temperature, no internal work is done, and the heat to be supplied,

$$Q = I + AU = 0 + ART \log_e r,$$

is just equal to the external work done. In the reverse operation of compression, work is done upon the gas, and heat must be abstracted from it in order to keep the temperature from rising.

A table of natural or hyperbolic logarithms, for use in this calculation, will be found in any good mechanical engineers' handbook. The relation to common logs is,

$$\log_e r = 2.3026 \log r. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

(*g*) A General Law of Expansion, which has important applications, is expressed by the equation

$$Pv^n = C, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

where the index n takes particular values for different cases.

With this curve,

$$\begin{aligned} U &= \int_{v_1}^{v_2} P dv = P_1 v_1^n \int_{v_1}^{v_2} \frac{dv}{v^n} \\ &= \frac{1}{1-n} P_1 v_1^n [v_2^{1-n} - v_1^{1-n}] \\ &= \frac{1}{n-1} [P_1 v_1^n v_1^{1-n} - P_2 v_2^n v_2^{1-n}] \\ &= \frac{P_1 v_1 - P_2 v_2}{n-1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (28) \end{aligned}$$

Putting this in terms of T, by Eq. (21), and reducing to heat units, we have

$$AU = \frac{AR(T_1 - T_2)}{n-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

(*h*) Adiabatic Expansion. — The most important case under the above law is the adiabatic* or no-transfer operation, in which no heat

* The word means, without passing through.

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is given to or taken from the gas as it expands or is compressed. This makes Q equal zero in Eq. (14), whence comes the characteristic relation

$$I + AU = 0, \quad \text{or} \quad AU = -\text{I} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

In expansion without heat supply, the gas does work at the expense of its internal energy and the temperature falls, the negative internal work, $c_v\,(T_2 - T_1)$, being just equal in amount to the positive external work. This gives the equation

$$\frac{AR\,(T_1 - T_2)}{n - 1} = c_v\,(T_2 - T_1): \quad . \quad . \quad . \quad . \quad (31)$$

and remembering that $AR = c_p - c_v$, Eq. (22), we have

$$c_p - c_v = (n - 1)\,c_v,$$

whence

$$n = \frac{c_p}{c_v} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

For this particular value of n, $0.2375 \div 0.1690 = 1.406$, the symbol k is used. Substituting k in Eq. (28) we get a definite formula for external work in terms of mechanical quantities; but generally the thermal formula,

$$AU = c_v\,(T_2 - T_1), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

is more convenient, since the change of temperature is an element of major importance in almost any problem that may arise, and hence will be calculated for its own sake.

(*i*) ADIABATIC TEMPERATURE RANGE. — To find the relation between the limiting temperatures in an adiabatic expansion from p_1v_1 to p_2v_2, we proceed as follows:

The adiabatic pressure-volume equation is

$$p_1v_1^{\,k} = p_2v_2^{\,k}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

Factor this into the form

$$p_1v_1 \times v_1^{\,k-1} = p_2v_2 \times v_2^{\,k-1};$$

then substitute from Eq. (21) and get

$$RT_1v_1^{\,k-1} = RT_2v_2^{\,k-1},$$

or

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1} = \left(\frac{v_1}{v_2}\right)^{0.406} \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

(*j*) THE ADIABATIC CURVE is drawn in Fig. 32 for a considerable range of expansion and of compression, with the isothermal dotted in, through a common point A, for comparison. Since the temperature drops in adiabatic expansion, the curve AD falls below the isothermal AF: at a given volume, as OT, the point N on the adiabatic curve represents a lower pressure in the gas than does the point R on the isothermal; and after expansion to a certain pressure, as OS, the gas under isothermal conditions fills a larger volume than does that expanded adiabatically.

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In compression, the external work done upon the gas is changed to heat and added to the internal energy, hence the temperature rises, and the curve AE goes above AG. Note the temperatures marked along the adiabatic curve ED.

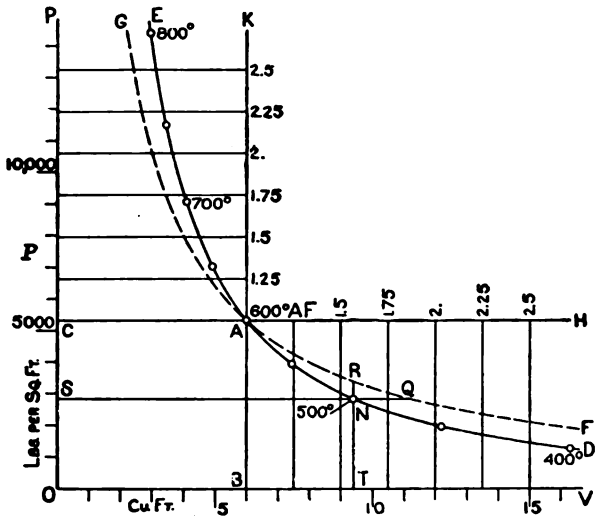

FIG. 32. — The Adiabatic Curve.

(*k*) PLOTTING THE ADIABATIC CURVE. — The task of drawing the adiabatic or any other curve of the form $pv^n = C$, if it involved the calculation of a set of coördinates directly from the equation, would

TABLE 3. FACTORS FOR THE ADIABATIC CURVE

1	2	3	1	2	3	1	2	3
Exp. $\frac{v}{v_1}$	Exp.	Comp.	2.25	0.3198	0.5617	8	0.0537	0.2274
			2.5	0.2757	0.5212	9	0.0455	0.2096
Comp. $\frac{p}{p_1}$	$\frac{p}{p_1}$	$\frac{v}{v_1}$	2.75	0.2412	0.4870	10	0.0393	0.1944
			3.0	0.2134	0.4578	12	0.0304	0.1708
			3.5	0.1718	0.4102	14	0.0245	0.1531
1.125	0.8476	0.9196	4	0.1424	0.3731	16	0.0203	0.1392
1.25	0.7307	0.8533	4.5	0.1207	0.3431	18	0.0172	0.1280
1.5	0.5655	0.7495	5	0.1041	0.3183	20	0.0148	0.1188
1.75	0.4553	0.6716	6	0.0805	0.2796	25	0.0108	0.1013
2.0	0.3774	0.6108	7	0.0648	0.2506	30	0.0084	0.0890

be rather laborious. To facilitate this operation, for the adiabatic of air, a series of ratios is given in Table 3, by means of which the ordinates can be found by a simple multiplication. In expansion, we take v as the independent variable, and locate a series of ordinates to the right

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of AB, on Fig. 32, at intervals each equal to one-fourth of the original volume v_1 or CA. Then multiplying the initial pressure AB or p_1 by the factors in the table, col. 2, we get the lengths of the successive ordinates. Thus in Fig. 32 as drawn, AB was 2.665 in., representing 5330 lb. per sq. ft., which is the pressure exerted when 1 lb. of air is confined in a space of 6 cu. ft. at 600 deg. absolute. The distances to be measured up from the base line on the successive ordinates were found as follows:

$\frac{v}{v_1}$ =	1.25	1.5	1.75	2.0	2.25	2.5
$\frac{p}{p_1}$ =	.7307	.5655	.4553	.3774	.3198	.2757
p =	1.947	1.507	1.214	1.006	0.852	0.734 in.

For compression, p is taken as the independent variable, and volume ratios from col. 3 of the table are used.

PROBLEMS

1. Let one pound of air at 72 deg. fahr. be heated up to 244 deg. under a constant pressure of 20 lb. per sq. in. absolute; find the initial and final volumes and the values of total heat, external work, and internal work. Get external work first in foot pounds, then in heat units.

2. A body of air which fills 5 cu. ft. at 100 deg. is heated to 250 deg. under a constant pressure of 60 lb. per sq. in. Calculate final volume, external work, total heat imparted, and internal work.

3. A cylinder 2 ft. in diameter and with the piston 1.5 ft. from the cylinder head is filled with air at 96 deg. fahr. and 4 atmospheres pressure; what work will be done by the air in isothermal expansion to three times the initial volume, and how much heat will have to be supplied?

First compute in terms of mechanical quantities, following Eq. (24); then find weight of air present and make a parallel calculation for external work by means of Eq. (25).

4. With the same initial conditions as in Problem 3, let the air be expanded adiabatically to the same final volume; what will be the final pressure, the final temperature, and the external work done?

Use Table 3 to get the final pressure. Having this, get external work by Eq. (28). After computing T_2, make another calculation of external work by Eq. (33).

5. Compute one of the factors in col. 2 of Table 3, say for an expansion ratio of 4.

6. Following the method of getting Eq. (35), find the similar relation between temperature ratio and pressure ratio.

7. If one pound of air, originally at 680 deg. fahr., and at 100 lb. per

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sq. in., is expanded adiabatically to a temperature of 40 deg., what will be the final pressure, the initial and final volumes, and the amount of external work done?

8. If 10 cu. ft. of air at 540 deg. fahr. and 125 lb. per sq. in. is expanded adiabatically till the pressure falls to 25 lb., what will be the final temperature and volume and the amount of external work done?

§ 8. The Ideal Heat Engine

We are now ready to develop the thermodynamic process of the ideal heat engine, which is to give the greatest possible efficiency in the conversion of heat into mechanical work, and serve as a standard of comparison for actual engines.

(*a*) THE THERMODYNAMIC CYCLE. — In every heat engine, the working substance goes through a circuit or cycle of operations. Starting at a particular condition, it passes through several changes of state, returning to the original condition. This continuity of process or closure of cycle, either actual or in equivalence, is essential if the working of the engine is to be continuous.

In general, the major divisions of the cycle are, an expansion during which heat is received from some source at high temperature and external work is done; and a compression during which external work is done upon, or received by, the working medium and heat is rejected to an outside body at low temperature. Each of these main divisions usually contains two distinct processes, and may contain more than two.

(*b*) POSITIVE AND NEGATIVE WORK. — Using the terminology of the ordinary cylinder and piston engine, the cycle consists of an out or working stroke and a return or compression stroke. Obviously, it is desirable that, for a given amount of heat supplied, the effective difference between the positive work of expansion and the negative work of compression shall be as large as possible, because the ratio of this effective work to the heat received measures the efficiency of the heat engine. Since the pressure of a given portion of gas, confined in a certain space, is proportional to the absolute temperature, and since the same range of volumes is passed through in both strokes, the evident requirement for high efficiency is that the expansion shall take place at the highest possible average temperature and the compression at the lowest possible average temperature.

(*c*) WORKING BETWEEN TEMPERATURE LIMITS. — The simplest conditions as to the supply and rejection of heat are, to have a source of heat maintained at a uniform high temperature and a heat absorber or cooler kept at a uniform low temperature. With constant temperature limits thus fixed, it appears that isothermal operations at the respective

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temperatures best meet the requirement just stated. But besides expansion at a constant high temperature and compression at a constant low temperature, there must be a drop from the high to the low temperature in one part of the cycle, and a return from the low to the high in another. For these operations, the adiabatic process, involving no transfer to or from the medium, naturally suggests itself.

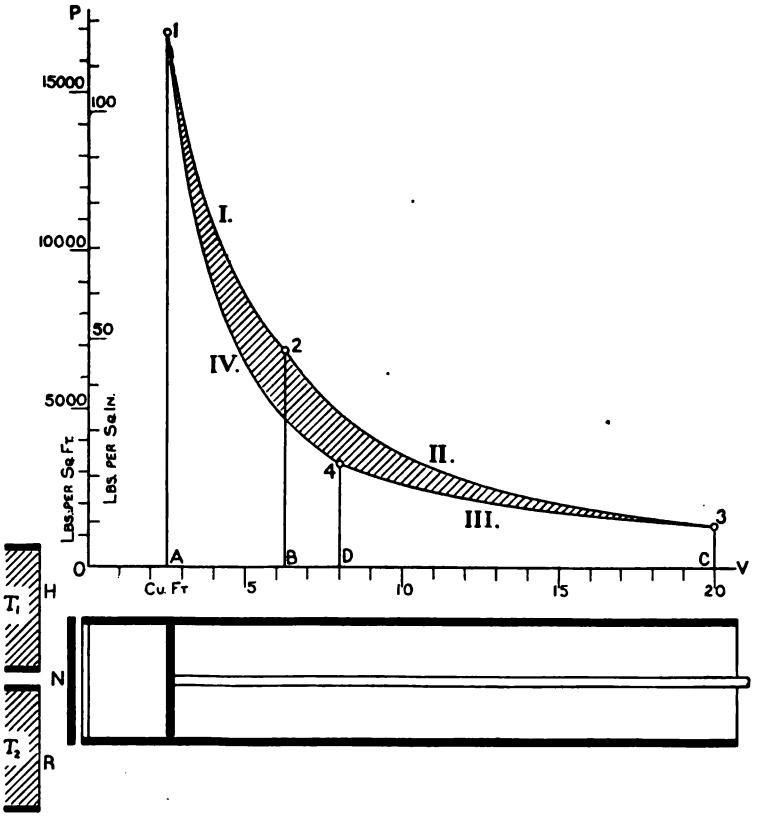

FIG. 33. — The Ideal Heat Engine.

(*d*) THE CARNOT CYCLE. — The French scientist Carnot first devised the cycle made up of two isothermals and two adiabatics. The essential apparatus is outlined and the diagram of the cycle given in Fig. 33. This ideal engine uses a confined body of air, which is alternately heated and cooled within the cylinder. All the surfaces in contact with the air must be thermally neutral, that is, must have zero capacity for heat, and all except the cylinder head must be perfectly nonconducting; the latter is a perfect conductor, but is provided with

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a removable cover N of the nonconducting material. These requirements as to material — very far from being realized in any actual engine — are imposed in order that adiabatic operations may be carried out. There is also a source of heat H, or heat reservoir at high temperature, and a heat receiver R at low temperature, for the isothermal operations. These are of very large capacity, so that they can give off and absorb heat without appreciable fluctuation in their own temperatures; and it is assumed that heat will pass freely to and from the air when the way is opened, without the temperature difference that is really necessary to flow of heat. In two respects then, as regards thermal physics, the scheme is entirely ideal, representing the unattainable limit toward which the actual processes can only approach. Mechanically, provision must be made for exerting a force along the piston rod always equal and opposite to the pressure of the air upon the piston, and for regulating the movement of the piston. To carry out the ideal of perfection, it is assumed that the machine operates without loss of work by friction.

(*e*) THE DESCRIPTION OF THE CYCLE shown in Fig. 33, with expressions for all the quantities involved, is as follows:

Start at point 1 of the diagram, with a unit weight of air under the conditions P_1, v_1, and T_1.

Phase I of cycle. Heater H on cylinder head, isothermal expansion at T_1, from 1 to 2: pressure-volume relation,

$$P_1 v_1 = P_2 v_2.$$

Heat received and work done,

$$Q_{\mathrm{I}} = AU_{\mathrm{I}} = ART_1 \log_e \frac{v_2}{v_1}.$$

Phase II. Cover N on cylinder head, adiabatic expansion, along 23, from T_1 to T_2:

$$P_2 v_2^k = P_3 v_3^k;$$

and from Eq. (35)

$$\frac{v_3}{v_2} = \left(\frac{T_1}{T_2}\right)^{\frac{1}{k-1}} \quad \ldots\ldots\ldots\ldots \quad (36)$$

Heat received,

$$Q_{\mathrm{II}} = 0.$$

Work done,

$$AU_{\mathrm{II}} = c_v (T_1 - T_2).$$

Phase III. Cooler R on cylinder head, isothermal compression at T_2, from 3 to 4:

$$P_3 v_3 = P_4 v_4.$$

Work received and heat rejected,

$$Q_{\mathrm{III}} = AU_{\mathrm{III}} = ART_2 \log_e \frac{v_3}{v_4}.$$

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Phase IV. Cover N on cylinder head, adiabatic compression, along 41, from T_2 to T_1:

$$P_4 v_4^k = P_1 v_1^k;$$

$$\frac{v_4}{v_1} = \left(\frac{T_1}{T_2}\right)^{\frac{1}{k-1}} \quad \ldots \ldots \ldots \quad (37)$$

Heat rejected,

$$Q_{IV} = 0.$$

Work received,

$$AU_{IV} = c_v (T_1 - T_2).$$

Now from Eqs. (36) and (37),

$$\frac{v_3}{v_2} = \frac{v_4}{v_1},$$

whence

$$\frac{v_3}{v_4} = \frac{v_2}{v_1} = r. \quad \ldots \ldots \ldots \quad (38)$$

That is, the two isothermal operations, between and limited by the same adiabatics, have the same ratio of volumes.

Now the external works in the two adiabatic phases, U_{II} and U_{IV}, are equal in amount and balance each other; the effective work is therefore

$$U = U_I - U_{III} = R \log_e r (T_1 - T_2);$$

and the ratio of this work to the heat received, or the efficiency of the engine, is

$$E = \frac{AU}{Q_I} = \frac{Q_I - Q_{III}}{Q_I} = \frac{T_1 - T_2}{T_1}. \quad \ldots \ldots \quad (39)$$

With this efficiency, the work gotten out of a given quantity Q of heat supplied is, using the factor J, Eq. (2),

$$U = JQ\frac{T_1 - T_2}{T_1}. \quad \ldots \ldots \ldots \quad (40)$$

(*f*) EFFICIENCY OF THE IDEAL ENGINE. — The result expressed by Eq. (39) is of the highest importance, reducing to quantitative terms the statement at the beginning of this chapter, that only a portion of any given supply of heat can be converted into mechanical work. The ideally perfect heat engine, working between the temperature limits T_1 and T_2 and receiving the heat supply Q, can at the best convert into work only the fraction $(T_1 - T_2)/T_1$ of this heat, and must unavoidably reject as heat the remaining portion, T_2/T_1. The engine with actual imperfections will convert less and reject more than these proportions.

It will be noted that while properties of the particular medium enter into the detail of the quantitative discussion in Art. (*e*), these all cancel out at the end. Partly from this fact, partly from lines of general

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thermodynamic reasoning which are set forth in § 10, the conclusion has been established that the efficiency here deduced is perfectly general, being independent of the medium and dependent only upon the realization of the isothermal and adiabatic operations.

In certain types of heat-engine plants (notably, the common gas engine with the explosive cycle), it is inherently impossible to supply heat to the working medium, or to reject heat, at anything like uniform temperatures. The ideal form of such cycles must be worked out in detail, for each case; but a rough measure of the limiting efficiency can be got by using the mean temperatures of supply and rejection in the ratio of Eq. (39) — this approximation being closer as the operations during supply and rejection are more nearly similar.

Before going on to develop more fully some general principles of the heat engine — which can be done in much better fashion after the temperature-entropy analysis is made available — we now show the utility of the ideal efficiency as a standard of comparison, in the following paragraph.

(*g*) Absolute and Relative Efficiency. — Suppose that a heat engine is found by test to receive a certain amount of heat Q and do an amount of work U. The actual ratio of heat converted to heat received,

$$E_A = \frac{AU}{Q}, \quad \ldots \ldots \ldots \quad (41)$$

is the absolute thermal efficiency of the engine. The efficiency of an ideally perfect engine working within the same limits, or the ideal efficiency, is

$$E_I = \frac{T_1 - T_2}{T_1}. \quad \ldots \ldots \ldots \quad (42)$$

The ratio of actual to ideal performance, or the relative efficiency,

$$E_R = \frac{E_A}{E_I}, \quad \ldots \ldots \ldots \quad (43)$$

is the true and proper measure of the effectiveness of the actual apparatus.

(*h*) Numerical Example. To show methods of calculation, and also in order to bring out certain properties of the Carnot cycle with a dry gas (not a vapor, like steam), the particular values belonging to the diagram shown in Fig. 33 will now be worked out.

One pound of air is the medium employed. The primary data are,

$T_1 = 800$ deg., $T_2 = 500$ deg., both absolute temperature;
$v_1 = 2.5$ cu. ft., $v_3 = 20$ cu. ft.

Then for phase I, $pv = 0.37\, T_1 = 296$,
and for phase III, $pv = 0.37\, T_2 = 185$.

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From these general expressions for the respective isothermal curves we get

$$p_1 = 296 \div 2.5 = 118.4 \text{ lb. per sq. in.},$$

and $$p_3 = 185 \div 20 = 9.25 \text{ lb. per sq. in.}$$

The adiabatic ratio is

$$r_a = \left(\frac{T_1}{T_2}\right)^{\frac{1}{k-1}} = (1.6)^{\frac{1}{0.406}} \qquad \log 1.6 = 0.20412$$

$$= 3.182 \qquad \frac{\log 1.6}{0.406} = 0.50276.$$

Now

$$v_2 = 20 \div 3.182 = 6.284, \qquad p_2 = 296 \div 6.284 = 47.10$$
$$v_4 = 2.5 \times 3.182 = 7.95, \qquad p_4 = 185 \div 7.95 = 23.27.$$

The two isothermal ratios come out as

$$\frac{v_2}{v_1} = \frac{6.284}{2.5} = 2.514 \quad \text{and} \quad \frac{v_3}{v_4} = \frac{20}{7.95} = 2.516.$$

Using $r = 2.515$, we get the work of phase I and the heat received as

$$U_I = 144\, p_1 v_1 \log_e r = 144 \times 295 \times 0.9223$$
$$= 39{,}179 \text{ ft. lb.};$$
$$Q_I = 50.36 \text{ B.t.u.}$$

With the temperature limits given, the efficiency is 0.375; therefore the net useful work done by the pound of air is

$$39{,}179 \times 0.375 = 14{,}692 \text{ ft. lb. or } 18.88 \text{ B.t.u.}$$

The external work of phase II is

$$c_v (T_1 - T_2) = 0.169 \times 300 = 50.70 \text{ B.t.u. or } 39{,}445 \text{ ft. lb.}$$

The important mechanical quantities for the whole cycle are now,

	Work	Mean pressure
Forward stroke	78,624 ft. lb.	31.20 lb. per sq. in.
Return stroke	63,932 ft. lb.	25.37 lb. per sq. in.
Net or effective	14,692 ft. lb.	5.83 lb. per sq. in.

The mean pressure is got by the method of Eq. (23), or by dividing work by piston displacement; the total piston displacement, from v_1 to v_3 is 17.5 cu. ft.; and in order to get pressures in pounds per square inch, the further divisor 144 must be introduced, so that the work quantities are here divided by $17.5 \times 144 = 2520$.

(*i*) AVAILABILITY OF THE CARNOT CYCLE. — These numerical results bring out the fact, also made evident graphically in Fig. 33 by the extreme vertical narrowness of the cycle diagram as compared with its height above the base line, that the net or useful forces are very small relative to the total forces acting. Consequently, this is a very

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poor cycle in the mechanical sense, entirely aside from difficulties in the way of its thermal realization. In order to embody this cycle, the engine must be very large, because of the small useful output per unit of cylinder volume, and at the same time very heavy in order to withstand the high pressures acting. Further, friction in the machine will be proportional to the big working forces which are so nearly self-balanced, not to the small net pressures; and if it bear even a moderate ratio to the former, it may absorb an overwhelming proportion of the useful work of the latter. For these reasons, the Carnot cycle is not available for practical application in the gas engine, but even as an underlying scheme of working must be greatly modified. With steam, however, as we shall see presently, it gives a much more effective and usable diagram, and with but little modification in intention, it is really the basis of the steam-engine cycle.

PROBLEMS

1. A heat engine working between the limits $T_1 = 1000$ deg. and $T_2 = 600$ deg. absolute receives 9200 B.t.u. per horse-power-hour; what is its absolute thermal efficiency and its relative efficiency, compared with the ideal engine?

2. A steam engine working between the limits $t_1 = 334$ deg., $t_2 = 112$ deg. fahr., is six-tenths as efficient as the ideal heat engine; how much heat must it receive per minute in order to develop 150 horse-power?

§ 9. The Temperature-Entropy Analysis

(*a*) THE IDEA OF ENTROPY. — All of the diagrams thus far used to represent thermodynamic operations — not including under this head the line of temperature relation on Fig. 28 — have shown the mechanical side of the subject, leaving the thermal side only implied or understood. The natural desire for an equivalent method of representing the operation directly in terms of heat, has led to the invention and development of the temperature-entropy diagram. As an obvious fact of nature, mechanical work is the product of two factors, either force and distance or pressure and volume; and if these are made the coördinates of a diagram, the area will show their product. For heat energy, temperature at once offers itself as one factor; but although specific heat is the other factor in ordinary calorimetry, it will not serve as the second coördinate of a thermodynamic diagram. Instead, we consider heat energy to be a compound quantity, take absolute temperature as one factor, and by division find the other factor, to which is given the name entropy.* There is no simple physical concept for this quantity,

* The Greek word means, "a turning toward": it may be paraphrased as "an impartation" or "something imparted."

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but it is not the mere mathematical abstraction that the definition just given might imply. It is best, however, to take as foundation the mathematical idea, which covers all use of entropy in diagram and calculation. A sense of reality and of fuller meaning will grow with use and familiarity.

When a portion of heat Q is imparted to a body at a temperature T, the body acquires the entropy N, measured by the quotient

$$N = \frac{Q}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (44)$$

If the operation takes place with varying temperature, so that only the infinitesimal dQ is imparted at any particular temperature T, we have

$$dN = \frac{dQ}{T}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (45)$$

whence, during the passage from T_1 to T_2, the entropy acquired is

$$N = \int_{T_1}^{T_2} \frac{dQ}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (46)$$

(*b*) THE MECHANICALLY SIMPLEST OPERATIONS first taken up in §§ 6 and 7 — namely, heating at constant pressure and at constant volume — both have the essential characteristic that the heat imparted bears a constant ratio to the change in temperature. Letting c stand for any constant specific heat, we have

$$dN = \frac{dQ}{T} = c\frac{dT}{T}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (47)$$

whence

$$N = c\int_{T_1}^{T_2} \frac{dT}{T} = c \log_e \frac{T_2}{T_1} = 2.3026\, c \log \frac{T_2}{T_1}, \quad . \quad . \quad . \quad (48)$$

the last expression being in terms of common logarithms.

For heating air at constant pressure, $c = 0.2375$, and Eq. (48) becomes

$$N = 0.54685 \log \frac{T_2}{T_1}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

while for constant volume the coefficient in Eq. (49) is

$$0.1690 \times 2.3026 = 0.38914. \quad . \quad . \quad . \quad . \quad . \quad (50)$$

An application of these formulas is seen in Figs. 34 and 35. The first shows the same operations as in Fig. 27, with the particular data that one pound of air is heated from 492 deg. to 984 deg. absolute in each case, so that T_2/T_1 is 2. Always, entropy is abscissa and temperature is ordinate. While the gas changes state from A to B, the entropy acquired is $0.5469 \times 0.30103 = 0.1646 = \text{OD}$; and from A to C it is $0.3891 \times 0.30103 = 0.1171 = \text{OE}$. Of course, calculation must

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be made for a number of intermediate points in order to lay out either curve on Fig. 35. The area under the curve equals the total heat imparted; thus ABDO = Q_p = 0.2375 × 492 = 116.85 B.t.u., and ACEO = Q_v = 0.169 × 492 = 83.15 B.t.u.

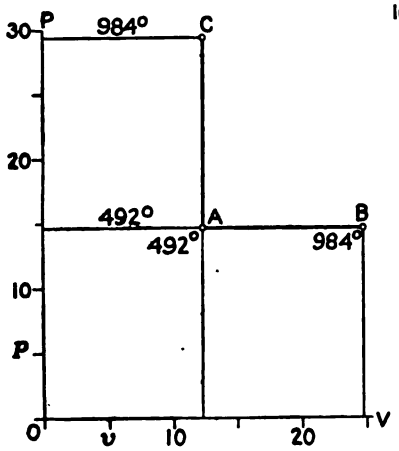

FIG. 34. — Mechanical Diagram for Heating at Constant Pressure and at Constant Volume.

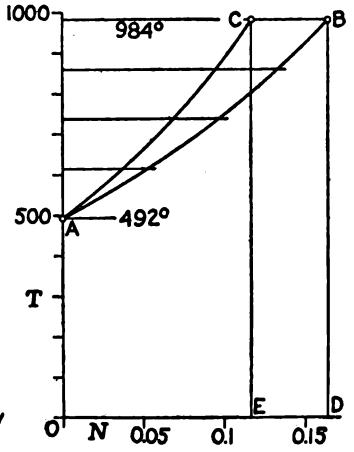

FIG. 35. — Thermal Curves for Operations in Fig. 34.

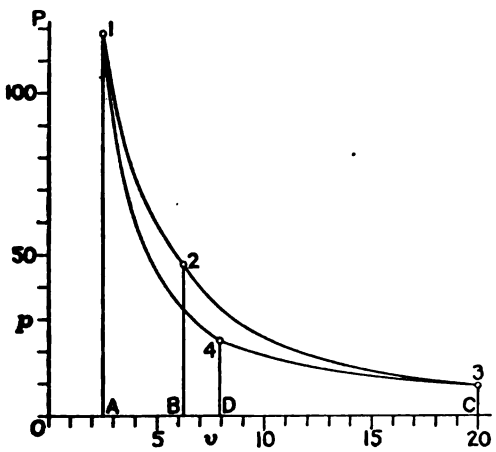

FIG. 36. — The Carnot Cycle as in Fig. 33.

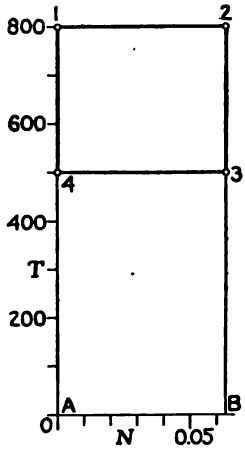

FIG. 37.— Temperature-Entropy Diagram for Fig. 36.

(c) ISOTHERMAL AND ADIABATIC OPERATIONS. — In the temperature-entropy system, or on the thermal side, the two simplest operations are the isothermal and the adiabatic, the elements of the Carnot cycle. The first is change of entropy at constant temperature, represented by

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a horizontal straight line on the diagram; the second is change of temperature at constant entropy, represented by a vertical line; and the TN diagram for the Carnot cycle is the rectangle shown in Fig. 37. In the analysis of a thermal operation, these are the ultimate components; for while other operations can be resolved into these components, they themselves cannot be reduced to anything simpler.

Equation (44) applies to Fig. 37; from 1 to 2 the entropy N = AB is acquired by the working medium, the heat received being $Q_1 = NT_1$ = area 12BA; the heat rejected in phase III is $Q_2 = NT_2$ = area 34AB, and the heat converted into work is

$$AU = N(T_1 - T_2) = \frac{Q_1}{T_1}(T_1 - T_2), \quad . \quad . \quad . \quad . \quad (51)$$

the same entropy being lost from 3 to 4 that was acquired from 1 to 2. Fig. 37 is drawn for the conditions of Fig. 33, which is partly reproduced in Fig. 36. The amount of heat received, Q_1 can be found only from the ratio of isothermal expansion, as in the example in § 8 (h), where it was calculated to be 50.36 B.t.u. for this particular case; then the entropy acquired, AB on Fig. 37, is got by plain division, its value being here $N = 50.36 \div 800 = 0.063$.

Having set forth the idea of entropy and briefly illustrated how it may be applied, we will now use it as a help in some further discussion of the Carnot cycle and of the heat engine in general.

PROBLEMS

1. In Fig. 35 is represented the heating of one pound of air from 492 deg. to 984 deg. absolute. For curve AB (or CD) find value of N at three intermediate points, at temperatures of 615 deg., 738 deg., and 861 deg.

2. If Fig. 39 (see forward for description of action represented) shows an operation with one pound of air, and the limiting temperatures are $T_1 = 800$ deg., $T_2 = 500$ deg. absolute, calculate the entropy values BD, BK, and BH.

§ 10. General Principles of the Heat Engine

(*a*) GENERAL THERMODYNAMIC IDEAS. — Heat energy has an active, continual, and irreversible tendency to drop from a higher to a lower intensity or temperature. If a hot body is placed in cooler surroundings, its intense heat energy tends to escape, diffuse itself, and settle to the level of the surroundings. When a body has been thus cooled by radiation and conduction, it still retains, at the lower level, the heat corresponding to that temperature.

Instead of simply escaping and diffusing itself, the intense heat energy — or we might say, the intense part of it — can be converted

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into mechanical work by means of a heat engine. Every such device takes in heat at high temperature and converts a portion of it; but there always remains a residuum, corresponding roughly to what is left in a body after simple cooling, which cannot be changed into anything else, but must be rejected as heat at low temperature.

The thermodynamic lowering of temperature, through the conversion of a portion of the supplied heat into work, is a useful process; cooling by diffusion is wasteful. For a given range of working, the Carnot engine is the most efficient possible because it takes in all its supply at the upper limit and rejects all the unused heat at the lower limit. If heat is available at a certain intensity, but is received into the working medium at a lower temperature (dropping across a temperature gap), there is evidently a loss of thermodynamic effect; similarly, if a part of the residual heat is rejected above the temperature of the receiver, all of the possible work has not been gotten out of it.

(*b*) THERMODYNAMIC AVAILABILITY. — Suppose that we have a heat quantity Q_1 capable of transfer at T_1, as, for instance, from the constant-temperature reservoir of the Carnot engine. Drawing on Fig. 38 the isothermal line AC and making AC or BD equal $N_1 = Q_1/T_1$, we have the energy Q_1 diagrammed in the rectangle ACDB; and at first we will say that the operation represented is simply the passage of the heat Q_1 from the source body, N_1 being the entropy given up by that body.

With an ideal heat engine, the line AC will also represent the reception of the heat Q_1 by the working medium; and if the cycle of the engine be performed, the heat area ACFE will be converted into mechanical work, while the remainder EFDB will be left as residual heat at T_2. For a certain rejection temperature T_2, the proportion of Q_1 that is available for conversion is greater as T_1 is greater or as N_1 is less. At the very least, the residual heat will be equal to N_1T_2.

In general, if a heat supply can be represented graphically, according to the temperature-entropy system — the diagram showing its possible manner of impartation by the source and of reception by the medium — only the area projecting above the level of the lowest temperature attainable in heat rejection can be transformed into work. In simplest statement, the thermodynamic value of a supply of heat depends upon the temperature at which it is available for reception by the heat engine (this to be considered in connection with the possible temperature of rejection).

(*c*) EFFICIENCY IN CONVERSION. — The absolute efficiency of any heat engine is greater as the heat rejected to the cooler is less. Assuming, for convenience (or as the ultimate result), isothermal rejection

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at T_2, and letting Q_2 stand for the heat rejected, we have $N_2 = Q_2/T_2$ as the entropy received by the cooler. Evidently, it is desirable to keep N_2 as small as possible.

In the ideal engine N_2 is equal to N_1; in an actual engine, it will be greater, appearing perhaps as BK on Fig. 38, where EJKB will then be the area of rejected heat. As the extreme opposite to ideal performance, let the heat Q_1 be, through some process, simply lowered from T_1 to T_2, as by plain cooling or "flow" of heat from a hot to a cool body: then the final entropy will be $N_0 = Q_1/T_2$, and area EGHB will be equal to ACDB in Fig. 38. If a heat-engine cycle, this process will be of zero efficiency, all of the available energy, area ACFE, being allowed to sink below the inferior limit of thermodynamic activity, the line EG.

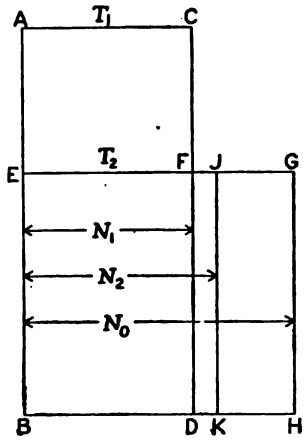

FIG. 38. — The Limits of Heat Conversion.

We see, then, that the limits of heat-engine efficiency are, zero on the one hand, with maximum increase of entropy from N_1 to N_0; and the ideal maximum on the other hand, with the cycle between constant entropy limits. An increase of entropy from heat reception to heat rejection, like FJ, measures the wastefulness of the actual engine, as compared with the ideally perfect cycle.

(*d*) THE RESIDUAL HEAT. — The general statement at the beginning of the chapter, that the whole of a given supply of heat cannot be converted into work, now takes the form that it is impossible to avoid the rejection, as heat at low temperature, of a considerable part of the heat received. The amount of this rejected heat we have just been considering; a fundamental argument for its unavoidability can be very simply expressed in terms of the entropy idea, as follows:

A body, such as the working medium in the engine, can only gain or lose entropy as it receives or gives up heat (not work). In a heat-engine cycle, the medium gains entropy with the heat received, and in order to return to the starting point it must get rid of this entropy. It can do so only by rejecting heat, and even if this be done at the lowest temperature, the amount so rejected will be at least N_1T_2.

(*e*) THE HYDRAULIC ANALOGY. — Perhaps the best physical idea of entropy can be got by comparing the heat-engine cycle with the operation of a water motor, which receives water at a level H_1 and discharges it at a level H_2 (both heights being measured above sea level

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as an absolute zero). The initial total potential energy of a weight W of water is WH_1; and at least the energy WH_2 will be left in it when it leaves the motor at what is, presumably, the lowest level of discharge permitted by the topographical location of the plant. Now the symbol W stands for both the force of gravity and the mass or substance quantity on which this force acts. We cannot see that entropy is analogous to force; but it does correspond quite well with the mass or substance of the water. As such an analogue, it may be thought of as a sort of energy vehicle, carrying heat energy into or out of a body; and the higher the temperature at which it passes from one body to another, the more energy does it carry. There is little profit in trying to press the analogy into the details of the respective weight-lowering and thermodynamic processes, but the general comparison gives to the concept of entropy about as much substantiality as it seems to be capable of possessing, in terms of the simpler ideas of mechanics.

(*f*) The Idea of Reversibility. — Having elaborated the principle of the Carnot cycle, showing more fully that it is the limit of heat-engine performance, our next step is to express in simplest yet most comprehensive terms that characteristic of the component operations of this cycle by virtue of which maximum conversion of heat into work is secured. This characteristic is completely defined by the term "reversible," but the special sense in which that word is now to be used must be made clear.

Consider first the purely mechanical case. To say that a machine is, in the present sense, reversible, does not mean merely that it can be run backward, but that all the force and work relations can be reversed. In a hoisting machine, let the force F be applied to lift a load W; then the load W can at any time become a driving force, running the machine backward against F as a resistance. For perfect reversibility, the force F must be the same in both directions of running; that is, the work by F and the work against W must be equal, or the machine must lose no work in frictional or other wasteful actions. Reversibility is therefore a criterion of mechanical perfection; if it were completely attainable, we should have what is called perpetual motion of the first order, in which a machine, once started and then let alone, would keep going forever.

This idea is somewhat limited in its applicability in the mechanical field, however, since there is a large class of machines which do their useful work against a resistance that cannot possibly move backward. A locomotive drawing a train on a level track works against the frictional resistances to the movement of the train; similarly, a form-changing machine, such as any machine tool, has a nonreversible resistance.

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But even if the concept cannot be formulated from the operation of these machines, it is of full effect as a criterion of perfection for them, implying as it does that no energy is to be lost within the machine itself.

(*g*) REVERSIBILITY OF PROCESS. — Now consider thermodynamic processes. On the mechanical side, any operation in which a pressure acts upon a movable confining surface, such as a piston, is reversible at this surface. If, for instance, a body of gas is expanded at constant pressure through a certain range and then compressed under the same condition through the same range, the amount of external work, positive or negative, will be the same in both cases. With friction in the machine, the work which it delivers during the expansion will be less than that which it must receive in order to perform the compression — but this is outside of the thermodynamic process.

Whether the heat received during an expansion can, in reversed performance of the operation, be put back where it came from, is a much larger question. Under Carnot engine conditions — see § 8 (*d*) — with the medium in thermal contact with the source, and with the assumption that no temperature difference is needed to make heat pass from one body to another, we can easily see that if an isothermal expansion be reversed, the heat will return to the source unchanged in quantity and state. Under adiabatic conditions, the energy that goes out of the medium as work during an expansion very evidently comes right back as heat during compression through the same range. If an operation with varying temperature and with heat interchange is to be reversible, it is necessary to have a heat reservoir with graded temperature — as, for example, a long pipe which is hot at one end and grows cooler toward the other — so that each bit of heat can be received or rejected at its own temperature.

Thermal reversibility implies, then, two conditions; first, that the medium shall be at the temperature of the source or receiver; second, that heat shall pass without a temperature difference to drive it. The first may be thermodynamically attainable (except as influenced by the second), but the second is physically impossible (just as is a machine without friction), although it stands as the limit of the attainable.

(*h*) AN IRREVERSIBLE CYCLE. — The subject may be clarified, perhaps, by discussing a cycle which is entirely lacking in thermal reversibility. Fig. 39 is, as concerns the outline ACFGHBA, exactly the same as Fig. 38, heat being supplied and carried away at the respective constant temperatures T_1 and T_2. Now let the gas start at E with the temperature T_2, and be heated at constant pressure up to T_1 at J; during this operation, each elementary quantity of heat except the very

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last will have to drop across the temperature gap from T_1 to the instantaneous (variable) temperature T. Reversing the operation mechanically, cool the gas under the same constant pressure, from J back to E, rejecting heat to the cooler across the gap from T to T_2. The net result is the mere transfer of a certain amount of heat from source to receiver, with no useful effect, realizing the zero-efficiency case of Fig. 38 and Art. (c). The irreversibility of the processes lies in the fact that it is impossible to make heat pass upward across a temperature gap, so that it cannot be put back where it came from. Of course, the areas ACDB, EJKB, and EGHB are equal.

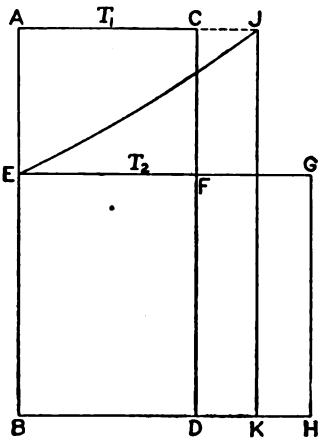

FIG. 39. — An Irreversible Cycle.

Note how the increase in entropy from source to receiver occurs. Because of the merely diffusive temperature lowering, entropy BK gained by the medium is greater than DB lost by the source; similarly, BH gained by the receiver is greater than KB lost by the medium.

(i) THE REVERSIBLE CYCLE. — To say that the Carnot cycle is reversible is in reality only another way of stating that all heat is taken in at the upper limit and rejected at the lower limit, but it is a short way of making the statement. Further, the Carnot cycle is a particular case, with the special requirement of isothermal transfer. The term reversible covers arrangements in which heat source and heat receiver are not of uniform temperature throughout the operations which they control, or in which the temperature limits are not the same for every element of the heat used; and in requiring that these limits be reached, whatever their form and however determined, reversibility becomes a characteristic of the ideal form of the cycle of any heat-engine plant.

(j) THE FULL ARGUMENT FROM REVERSIBILITY, to establish the Carnot cycle as the limit of thermodynamic performance — and, by inference, any other reversible cycle, with its own particular temperature ranges — is as follows:

Suppose that with the apparatus illustrated in Fig. 33 the cycle is performed backward, in the order 14321. During the isothermal expansion 43 the heat quantity R_1 will be drawn from the cold body at T_2; to this will be added the effective work AU_1 of the cycle, and the resulting heat, $H_1 = R_1 + AU_1$ will be rejected into the hot body at

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T_1.* The essential fact is that H_1 and R_1 are respectively the same in amount whether the heat engine works directly or reversed.

Now to drive this reversed engine or heat pump, we employ another heat engine, connected to the same source and receiver, drawing from the former the heat H_2 and rejecting to the latter the heat R_2. If we assume that there are no losses of mechanical work, as by friction, the second cycle will develop just the work required to drive the first backward, and the whole combination will embody a self-contained process for converting heat into work and then converting it back again. We wish to prove that the second engine, of which the manner of operation may be anything imaginable within the limits imposed by source and receiver, cannot possibly be more efficient than the first reversible engine.

Since the work quantities are the same

$$H_2 - R_2 = H_1 - R_1. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (52)$$

In the first engine, as stated above, the heat quantities H_1 and R_1 bear the same relation to each other and to the work of the cycle in reversed as in direct operation; therefore the efficiency when working as an engine is

$$E_1 = \frac{H_1 - R_1}{H_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (53)$$

And the efficiency of engine number two must be

$$E_2 = \frac{H_2 - R_2}{H_2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (54)$$

Now if it were possible for the second engine to be more efficient than the first, we should have

$$\frac{H_2 - R_2}{H_2} > \frac{H_1 - R_1}{H_1}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (55)$$

this can be true only if H_2 is less than H_1, since the numerators of the fractions are equal. If H_2 were less than H_1, and R_1 greater than R_2, or if less heat were taken from the high-temperature source than was rejected into it, and if more heat were taken from the low-temperature receiver than was given to it, there would be a net effect of transferring heat from a cold body to a hot body by means of a self-contained process.† According to all our knowledge of the laws of nature, this is

* This scheme of working finds its practical exemplification in the refrigerating machine.

† The statement of the impossibility of this operation is commonly called the second law of thermodynamics; in final and most general terms it expresses and accounts for the fact that a heat engine can convert into work only a portion of the heat which it receives. The first law is the introductory statement of this chapter, that heat and work are interconvertible, in definite ratio.

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absolutely impossible; and we must conclude that the reversible heat cycle represents the limit of attainment, toward which the actual engine may approach, although it can never reach this limit.

This argument is, of course, a parallel to the train of ideas summed up in Art. (*d*). A self-propelled transfer of heat from a cold to a hot body and a decrease of entropy from source to receiver in a cycle would be equivalent effects, either one implying what is called perpetual motion of the second order, which means that more energy would be got out of a process than was put into it. This is decidedly more impossible than merely to pass energy through a process without any diversion from the main channel, which is what the ideal heat engine and the frictionless machine are required to do.

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CHAPTER III

PROPERTIES AND BEHAVIOR OF STEAM

§ 11. Generation and Properties of Steam

(*a*) The Generation of Steam. — The operation of making steam, as carried out in the ordinary steam boiler (and regarded from the viewpoint of thermal effect), is made up of two parts. The water, at some initial temperature t_0, is pumped into the boiler against the internal pressure p; and the first step is the heating of this water up to the boiling point t, or to the temperature of steam formation which corresponds to the pressure p. The boiling point having been reached, evaporation begins; and the water gradually turns into steam, with a very great absorption of heat and increase of volume, but with no change of temperature so long as the pressure is kept constant by permitting a continual discharge of steam, as to an engine. Coming thus from the water and in the condition in which it must exist while in the presence of water, the steam is said to be saturated, or it is a saturated vapor. Conducted away from the water and receiving more heat, with rise of temperature, the steam becomes superheated and approaches a gas in properties and behavior. Superheating, when performed, is to be considered as a third stage in steam formation, following after water heating and vaporization.

(*b*) Quantitative Values and Relations. — The laws governing the perfect gas are rational in character, logically deducible from a few fundamental experiments. The corresponding relations for water and steam are not thus rational but are almost wholly empirical, in the sense that the numerical values of the various quantities had to be found by means of a great number of careful experiments, extending over the range of variant conditions. The results of these experiments are plotted, compared, and combined, and from them are derived laws and relations which may or may not be capable of exact mathematical formulation. In any case, the formulas are too complicated for ordinary use, and for all practical purposes we must depend upon tables and diagrams.

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(c) STEAM TABLES. — Numerical and graphical tables of the properties of steam are given in the Appendix, pages 578 to 615. The descriptive list on page 573 defines the ground covered by the several tables; the succeeding list of symbols and quantities on pages 574 to 577 is convenient for immediate reference from the tables, and serves to direct further reference back to the descriptions and discussions in the present chapter.

The most general tables, numbers II and III, are carried up to the "critical temperature," 689 deg. fahr., above which it is impossible for water to exist in the liquid state. To cover the farthest range of attempted or exceptional use of high-pressure steam, the closely spaced part of Table II runs up to 550 deg. or to about 1000 lb. pressure; the limit of ordinary technical practice is at about 300 lb. or 420 deg. For superheated steam there is a similar liberal extension beyond the limits of ordinary practice, Table Diagrams VII and VIII running to 1200 deg., while steam is seldom raised above 700 deg. for use in engine or turbine.

Full descriptions of the quantities tabulated and diagrammed will be found in the next two sections. A presentation and discussion of the sources of the steam table is beyond the scope of this book, but a few notes and the more important references are given in the last section of the Appendix.

§ 12. The Pressure and Volume of Steam

(a) THE PRESSURE-TEMPERATURE RELATION. — For saturated steam, or for the process of vaporization, there is a single, definite relation between pressure and temperature, instead of the more general relation among pressure, volume, and temperature that holds for a gas. If water is subjected to a certain pressure p, it will not boil until the temperature t corresponding to that temperature is reached; and during vaporization both water and steam remain at this temperature. The relation between p and t is given numerical form in Table I and in the first two columns of Table II, and is shown graphically in the curves A1, A2, and A3 on Fig. 40. At low temperatures the pressure is small and varies slowly, at high temperatures it is large and varies rapidly; and with a total range from less than 0.1 lb. per sq. in. to nearly 3000 lb., there is a real necessity for breaking the pt curve into several sections, with different scales for p, as is done in Fig. 40.

As the temperature of steam falls below 212 deg., its pressure drops farther and farther below that of the atmosphere: this fact underlies the action of the steam-engine condenser, which is used to diminish the back pressure of exhaust steam.

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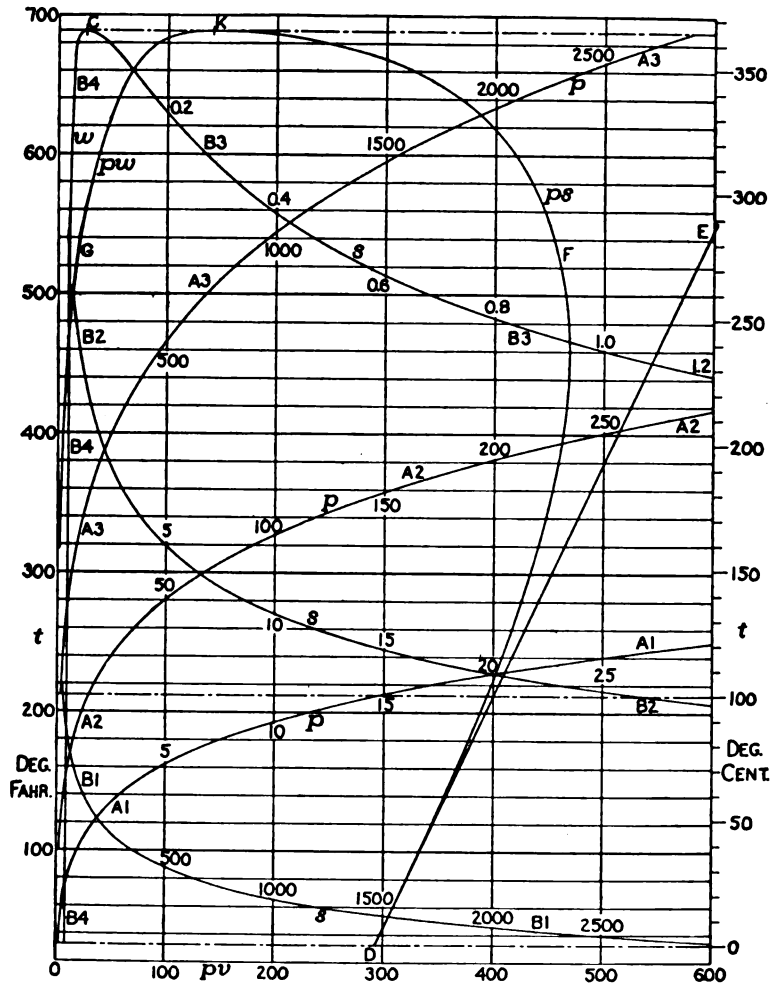

FIG. 40. — Pressure and Volume Curves for Saturated Steam.

Vertical base, temperature fahrenheit, with the centigrade scale added at the right.

Curves A1, A2, A3 show the saturation pressure p, plotted from col. 1 of Table II; this is really one curve, but for clearness must be thus broken into sections, with a different scale for each as marked along the curve: § 12 (*a*).

Curves B1, B2, B3 show in similar fashion the specific steam volume s, col. 2 of Table II. To the same scale as B3, the water volume w, Table III, col. 2, is laid out in B4, the two curves meeting in a rounded peak at C, at the critical temperature: § 12 (*c*).

The continuous curve DFKG is a plot of the pressure-volume product, ps for steam, pw for water, from cols. 4 and 5 of Table III, to the scale at the bottom of the diagram. With this, line DE shows the "ideal" product ps' or RT: § 12 (*e*).

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Most of the properties of steam vary in simpler fashion if developed on saturation temperature as a base, rather than on pressure: for this reason t is the "argument" or independent variable in Table II, the principal steam table. In practical observation of steam-plant performance, however, pressure is commonly the primary quantity; and to facilitate the use of Table II, Table I gives the values of t corresponding to equally spaced values of p. The necessity here met of a frequent change in the p interval shows the chief disadvantage of p as a basal quantity for the general table. The fact that a saturation-temperature base crowds together the high pressure range (where thermal conditions vary slowly with pressure), and spreads out the low-pressure range (where they vary rapidly), is the reason for making this temperature the determinant in diagrams where pressure is distinctly the fundamental quantity. This remark applies especially to the curves in Figs. 46 and 48 and in Table VIII, each of which is for a particular (constant) value of p and extends out from saturation into the region of superheat, and which are nevertheless located by starting points at ten-degree positions on the saturation line.

(*b*) Measurement of Pressure. — Up to this point we have taken p to be absolute pressure, measured above zero or from a state of perfect vacuum (and shall continue to do so unless otherwise stated). All ordinary pressure gages, however, measure from atmosphere as a starting point, whether upward as on a boiler or downward as on a condenser.

Table 4. Pressure-unit Ratios.

	Lb. per sq. in.	Lb. per sq. ft.	Atmosphere	Inches merc.	Mm. merc.	Kg. per sq. cm.	Kg. per sq. m.
1 lb. per sq. in. =	1.00	144	.06804	2.036	51.71	.07031	703.1
1 lb. per sq. ft. =	.00694	1.00	.....	.0141	0.359	.....	4.8825
1 atmo. =	14.697	2116.4	1.00	29.92	760	1.0334	10.334
1 inch merc. =	0.4912	66.33	.03342	1.00	25.40	.00345	34.54
1 mm. merc. =	.01934	2.785	.....	.0394	1.00	.00136	13.60
1 kg. per sq. cm. =	14.223	2048.1	0.9677	28.95	735.4	1.00	10,000
1 kg. per sq. m. =	.00142	0.205	.....	.0029	.07354	0.001	1.00

To use this table, take the name of the first unit in the column at the left and run along its line till the value in terms of the other unit is reached, in the column headed by the name of that unit. These ratios can be selected in pairs which are reciprocals of each other; the values left blank are reciprocals of large numbers, and would require too much space because so many zeros precede the significant figures.

To their indications must be added either the normal atmosphere, 14.7 lbs. per sq. in., or, in more precise work, the actual barometric reading at the particular time and place. Numerically, the reading of a vacuum

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gage is subtracted from the reading of the barometer in order to get the absolute pressure in the condenser. Vacuum gages, from their analogy to the barometer in range and because of the former common use of the mercury manometer in this service, are generally graduated in inches of mercury column.

In the metric-centigrade countries, the common unit of pressure is the kilogram per square centimeter; while for work calculation kilograms per square meter correspond to pounds per square inch. Ratios of equivalence among the various pressure units are given in Table 4.

(*c*) SPECIFIC VOLUME AND DENSITY. — For volume we shall use three symbols; in general, v will represent the volume of one pound of substance in cubic feet, under any conditions; in particular, w will be the volume of one pound of water and s the volume of one pound of saturated steam. For density, which is the reciprocal of specific volume, showing weight per cubic foot, we shall use d_w for water and d_s for saturated steam. The volume and density of water are given in Table III, cols. 2 and 3. Steam volume s fills col. 2 of Table II, but d_s is not tabulated. In Fig. 40, s is plotted after the same manner as p, in the curves marked B; and to the very large scale of the upper section B3 is laid out also the water volume w in curve B4, these two curves meeting in a rounded peak at the critical "point" C.

(*d*) VOLUME IN PARTIAL VAPORIZATION. — As indicated in § 11 (*a*), the process of vaporization (under constant pressure) begins with one pound of water at t and p and ends with one pound of steam at the same temperature and pressure. During the operation the volume changes from w to s; for the increase of volume we use the symbol u, so that

$$u = s - w. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (56)$$

The proportion of the volume change u that has been effected at any point in the process is the same as the proportion of the pound of water that has been turned into steam. If x be the proportion or fraction of steam (by weight) at any instant, and $(1 - x)$ or m the fraction of water or moisture, the volume will be either

$$v = w + xu, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (57)$$

or

$$v = s - mu. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (58)$$

In many cases, when x is large or m small, it is quite accurate enough to use

$$v = xs. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (59)$$

This is especially appropriate at low or moderate pressures, where the difference between u and s is relatively insignificant.

EXAMPLE 1. — If one pound of water be 60 per cent vaporized at a pressure of 100 lb. absolute, what space will it occupy?

By Table I the temperature is 327.86 deg. From Table III, w is 0.0177

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or 0.018, and from Table II, s is 4.430; then u is 4.412, and for v we have either

$$v = 0.018 + 0.6 \times 4.412 = 0.018 + 2.647 = 2.665 \text{ cu. ft.},$$

or

$$v = 4.430 - 0.4 \times 4.412 = 4.430 - 1.765 = 2.665 \text{ cu. ft.}$$

(*e*) THE PRESSURE-VOLUME PRODUCT. — More useful than the simple plot of volume on temperature (curve B1-B2-B3, Fig. 40), as a help in studying the properties of steam and to show how actual steam differs from a perfect gas in behavior, is the diagram got by plotting the product pv on t. This also is shown in Fig. 40, ps by the curve DFK, pw by GK. Values of these products are given in Table III, cols. 4 and 5. The straight line DE shows how the product pv would vary with t if the steam were a perfect gas. It represents the equation

$$pv = RT, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (60)$$

and for H_2O gas, on the basis of molecular weight, the value of R would be 0.5956 — compare 0.37 for air, Eq. (12).

(*f*) THE SATURATION LINE. — The curve showing the volumetric condition of steam which is completely vaporized but not at all superheated, whether s on t, s on p, or ps on t or p, is called the saturation line. When approached from the side of a steam-and-water mixture or of wet steam, it might better be called the dry-steam curve. As the limit of any operation of cooling from a superheated condition, showing where further abstraction of heat will begin to cause condensation, it is closely described by the term saturation line. In any case, the curve stands out as a boundary line between these two general states.

(*g*) THE IDEAL VOLUME OF STEAM. — As steam is more highly superheated, it comes nearer and nearer to the simple law expressed by Eq. (60): but toward saturation there is a shrinkage of volume or a decrease in the value of pv, shown in its full effect by the inward bending of the curve DF from the straight line DE on Fig. 40. This shrinkage is inappreciable at very low pressures, but becomes greater as the pressure is higher and the steam more dense. The molecules being crowded closer together, their inter-attractions become of considerable magnitude; and it is thought that even before the point of actual condensation is reached, the molecules may begin to coalesce in small groups.

By evaluating Eq. (60) for the coincident pressure and temperature of saturation, we get the "ideal" value of s for these conditions: the operation is,

$$s' = \frac{0.5956}{p}(t + 459.6)^*. \quad . \quad . \quad . \quad . \quad . \quad . \quad (61)$$

* This more precise value is used in making calculations for the tables, instead of the even 460 deg.

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The results of this computation are given in col. 3 of Table II — not for illustrative purposes, but as a help in getting the volume of superheated steam at any pressure and temperature. The difference $(s' - s)$ ranges from 1.3 cu. ft. at 32 deg. to 0.145 cu. ft. at 550 deg.; but while its absolute value is small and decreasing, its relative importance increases rapidly, as appears from the following figures, namely, that at 32 deg. the difference is 0.04 per cent of s, at 212 deg. it is 1.65 per cent, at 400 deg. 10.8 per cent, and at 550 deg. 34 per cent.

(*h*) THE CHARACTERISTIC EQUATION. — The general pressure-volume-temperature equation for steam at and beyond the state of complete vaporization is,

$$pv = RT - F_p f_t, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (62)$$

where F_p and f_t are functions of pressure and of temperature respectively, and their product serves as a corrective term, to change from the ideal to the actual value of pv. These factors are determined by conditions at the saturation line, as follows:

For the various temperatures in col. 0 of Table III, values of RT or of $[0.5956\ (t + 459.6)]$ were computed, and from them were subtracted the values of ps given in col. 5, to get $(RT - ps)$, which is equal to $F_p f_t$. Following Linde — see note 4, page 616, Appendix — the pressure factor F_p is taken as

$$F_p = p + \frac{p^2}{700}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (63)$$

of which the calculated values are not printed. Then division of $(RT - ps)$ by F_p and interpolation give f_t as in col. 5 of Table II. Above 535 deg. there is a change in method, f_t being assumed and F_p. derived from it by the same operation of division. High-range values of f_t, from 600 deg. to 1300 deg. fahr., are given in Table IV.

(*i*) THE VOLUME OF SUPERHEATED STEAM. — Cols. 3, 4, and 5 of Table II contain numbers to be used in getting the volume of steam at any pressure and temperature, by a method more convenient for regular use than the direct evaluation of Eq. (62). First, divide both members of Eq. (62) by p; the resulting expression for v is

$$v = \frac{R}{p} T - \frac{F_p}{p} f_t = v' - f_p f_t. \quad . \quad . \quad . \quad . \quad . \quad . \quad (64)$$

Here v' is the ideal volume, under Eq. (60), and the product $f_p f_t$ diminishes it to the actual volume. The factor f_p is, from Eq. (63),

$$f_p = \frac{F_p}{p} = 1 + \frac{p}{700}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (65)$$

it is not tabulated — except in Table V, above the range of this for-

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mula — because to write it out from the formula is at least as easy as to interpolate in a table which has temperature as argument.

It is best to think of volume v as reached by a constant-pressure expansion from saturation temperature t_s (plain t in the tables) to existing temperature t; then the easiest way to get the ideal volume v' is to take it as

$$v' = s' + \frac{R}{p}(t - t_s) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (66)$$

This is preferable to the operation

$$v' = \frac{R}{p}(t + 459.6) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (67)$$

because, with the usual range of not very great superheat, the quantity $(v' - s') = \frac{R}{p}(t - t_s)$ can be computed closely enough with the slide rule, but in order to get the whole of v' from T we must use more precise methods of arithmetic.

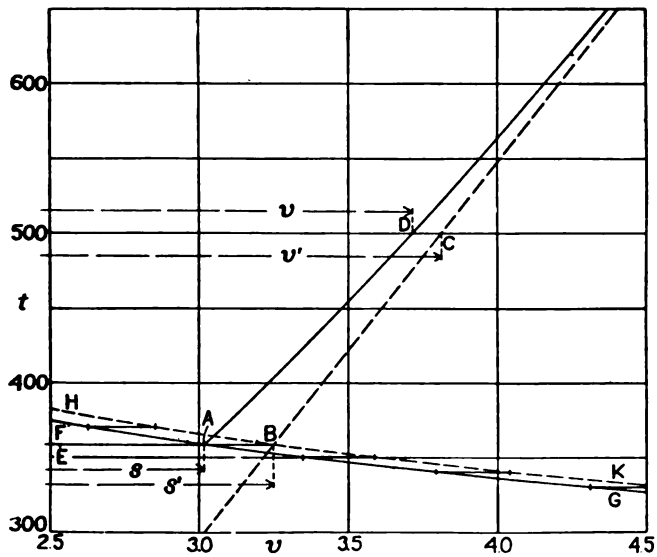

FIG. 41. — Method of Calculating the Specific Volume of Superheated Steam.

The scheme of this calculation is represented in Fig. 41. Saturation volume s is plotted in curve FG, which is therefore the saturation line, and the ideal volume s' in curve HK. At 150 lb. pressure — assuming conditions as in Example 2 — AB is the value of $(s' - s)$, or of $f_p f_t$ at saturation temperature. Having s' at B, we compute $(v' - s')$

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in order to get v' at C, then subtract $f_p f_t$ as computed for the temperature at line CD.

With the high-range values of f_p in Table V is given a quantity called Δf_p; this is the difference between f_p as tabulated and the value got from Eq. (65) for the determining pressure. To get intermediate values of f_p, interpolate Δf_p and add it to the number given by Eq. (65). Thus at 980 lbs. we should have 2.4000 from the formula and 0.0029 for Δf_p, giving 2.4029 for f_p. If a high degree of consistency is desired a curve of Δf_p can be plotted, instead of using plain, straight-line interpolation.

EXAMPLE 2. — Find the volume of one pound of steam at 150 lb. absolute and at 500 deg. fahr.

For $p = 150$, $t_s = 358.5$, from Table I; then by Table II, $s = 3.616$, $s' = 3.249$, $R/p = 0.003970$, $f_p = 1.214$; at 500 deg., $f_t = 0.080$, $(t - t_s) = 141.5$.

$$s' = 3.249 \qquad v' = 3.811$$
$$\frac{R}{p}(t - t_s) = 0.562 \qquad f_p f_t = 0.097$$
$$v' = \overline{3.811} \qquad v = \overline{3.714}$$

In Table VI this volume can be read as 3.71 cu. ft., which would be quite close enough for most purposes; the disadvantage of that diagram is the impossibility of close interpolation on pressure lines other than those which are drawn in and divided.

(*j*) EXPANSION UNDER CONSTANT PRESSURE. — Fig. 41 shows the change of volume with temperature under constant pressure. Coming down from high superheat, the ideal line CB would keep straight on till it crossed the vertical axis of temperature, here five divisions beyond the left edge of the diagram, at absolute zero. The actual line DA swings inward, v contracting more rapidly than v', until it gets to saturation at A; then, with condensation taking place, v contracts along the isothermal line AE to the water volume w.

The rate of change of volume with temperature under constant pressure, defined mathematically as $\left(\frac{dv}{dt}\right)_p$, is of interest. For the ideal gas, as readily appears from Eq. (60), it is $\frac{R}{p}$; for actual steam it is, from Eq. (64),

$$\left(\frac{dv}{dt}\right)_p = \frac{R}{p} - f_p \frac{df_t}{dt}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (68)$$

The last term is the rate of change in f_t per degree; it is negative, so that the term is really additive, and it can be accurately enough got from Table II.

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EXAMPLE 3. — For the conditions of Fig. 41 and Example 2, find the rate of expansion at 500 deg., and at saturation. Also, find where tangents to the curve AD will cut the temperature axis in each case.

In passing 500 deg., f_t changes from 0.0827 at 495 deg. to 0.0775 at 505 deg., or at the rate of -0.00052 per degree; multiplying this by f_p or 1.21, we get 0.00063 to be added to R/p or to 0.00397, making the rate of change 0.00460 cu. ft. per degree. At 358.5 deg., the rate of change in f_t is -0.00112, and the rate of expansion is $0.00397 + 0.00124 = 0.00521$.

Now at 500 deg., the volume of the pound of steam is, from Example 2, 3.714 cu. ft.; to reduce this to zero at the rate of 0.00460 per degree would require a drop of $3.714 \div 0.0046 = 807.4$ degrees. The absolute temperature at D, Fig. 41, is 959.6 deg., so that a line tangent to the curve AD at D will cross the axis of temperature or the line of zero volume at $959.6 - 807.4 = 152.2$ deg. absolute. Similarly, at 358.5 deg., $3.016 \div 0.00521 = 578.9$, the absolute temperature is 818.1 deg., and the tangent crosses at $818.1 - 578.9 = 249.2$ deg. absolute.

(*k*) VARIATION IN RATE OF EXPANSION. — The example just worked clearly exhibits the fact that the rate of expansion under constant pressure is greater near saturation than farther out. The inverse of this, that the rise of temperature required for a certain increment of volume is less near saturation, is equally clear on Table-diagram VI. The divisions on the horizontal lines of constant pressure mark off temperature intervals corresponding to equal changes of volume. Near saturation, the intervals are appreciably shorter, especially at high pressures.

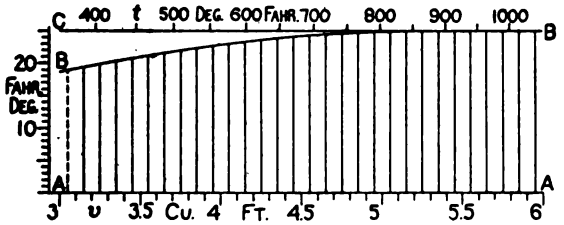

FIG. 42. — Temperature Intervals for Volume Increments of 0.1 cu. ft., at 150 lb. Pressure.

To show this up more strongly, the diagram in Fig. 42 is plotted, for the pressure used in the last two examples. The ordinate between base AA and curve BB is the number of degrees required to change the volume by 0.1 cu. ft. These ordinates are located, with reference to the actual volume, by the scale along the bottom, each at the middle of the increment which it produces; and at the top is a variant scale of corresponding temperatures. The height from AA to CB is the value (constant) which the temperature interval would have if the steam followed the law of Eq. (60): the actual intervals approach this limit,

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becoming indistinguishable from it, on the diagram, at high temperatures. To get AC, divide the volume increment 0.1 by the ideal rate of expansion, R/p; then

$$\text{AC} = 0.1 \times \frac{p}{R} = \frac{0.1 \times 150}{0.5956} = 25.02 \text{ deg.}$$

(*l*) To FIND THE TEMPERATURE which, under a certain pressure p, corresponds to a particular volume v, is a troublesome operation with the data in Table II; since f_t depends upon the final, unknown temperature, a series of trial solutions is necessary, each based upon the one ahead of it and coming closer to the true result. Table VI gives just this quantity, however, and if the precision with which it can be read is insufficient, at least furnishes a close approximation upon which to base a single corrective calculation. An example will illustrate the procedure to be followed.

EXAMPLE 4. — Under a pressure of 180 lb., find the temperature at which one pound of steam will fill a space of 3 cu. ft.

On the 180-lb. line, Table VI, the reading for 3 cu. ft. is about 481 deg. The saturation temperature being 373.15 deg., Table II gives $s' = 2.755$, $R/p = 0.003309$; at 481 deg., $t - t_s = 107.8$, and $f_t = 0.0903$; at 180 lb., $f_p = 1.257$. Now making the same calculation as in Example 2, we have

$$\begin{aligned} s' &= 2.755 & v' &= 3.1117 \\ \frac{R}{p}(t - t_s) &= 0.3567 & f_p f_t &= 0.1134 \\ v' &= \overline{3.1117} & v &= \overline{2.9983} \end{aligned}$$

This makes the volume 0.0017 cu. ft. less than the desired 3 cu. ft. With R/p equal to 0.00331, the needed increase will be produced by a rise of $0.0017 \div 0.0033 = 0.5$ deg., so that the exact temperature sought is 481.5 deg. instead of 481 deg. It must be noted that because f_t changes with t, this last simple correction can be made over but a very short distance; if the difference between t from Table VI and t as just computed had been even as great as one or two degrees, a second calculation would be necessary in order to fix t within 0.1 deg.

(*m*) HEATING AT CONSTANT VOLUME. — Sample curves for this operation with one pound of superheated steam, in a simple plot of p on t, are given in Fig. 43, AB for 3 cu. ft., GH for 24 cu. ft. The straight line EF represents the ideal line of pressure variation, related to AB, which would be traced if the law $pv = RT$ were in control. The saturation line SS is not only the lower limit of a set of curves like BA, but each of these curves simply turns a corner at its A point, and all run down together along the saturation curve. That is, if cooling at constant volume be carried into the region of wet steam, there will be an increasing proportion of condensation as the temperature falls, but the pressure (now independent of volume) will merely drop according

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to the saturation relation. A quantitative illustration of this progressive condensation will be found in § 13 (*m*) and Fig. 48.

The short plotted lines of points on Fig. 43 show the experiments of Linde and associates — see Note 4, page 616, Appendix — which are decidedly the best measurements of steam volume related to pressure

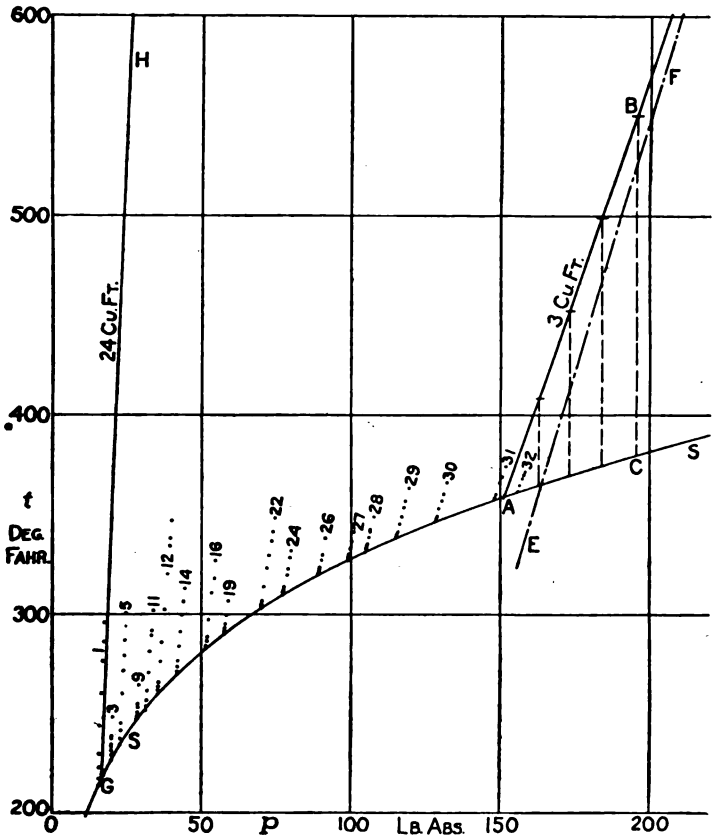

FIG. 43. — Curves of Heating at Constant Volume, for Superheated Steam.

that have as yet been made. In these, pressure and temperature were varied together, while volume was kept constant and weight of steam present was known. On these observations was based Linde's formula for the relation among p, v, and t, from which Eq. (62) as here evaluated departs by only a very small numerical difference. The purpose in plotting the experiments is to show how closely they agree in form with the curves got from the formula. It appears that the observations do not run far into the region of superheat, neither do they go to very high pressures.

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(*n*) THE ISOTHERMAL CURVE. — In Fig. 44 the isothermal is plotted in two ways, on pressure p as base (vertical). The curves in group I show product pv, while those in group II show volume directly. The same letters are used on corresponding points of the two groups. The ideal product RT or ps' from Eq. (60) now becomes the curve AC, instead of the straight line DE on the temperature base of Fig. 40, and AB is the saturation line. The ideal isothermal $pv' = C$ is represented by the vertical straight line CD; the actual isothermal BD is an arc of a parabola, as appears from the form of F_p in Eq. (63).

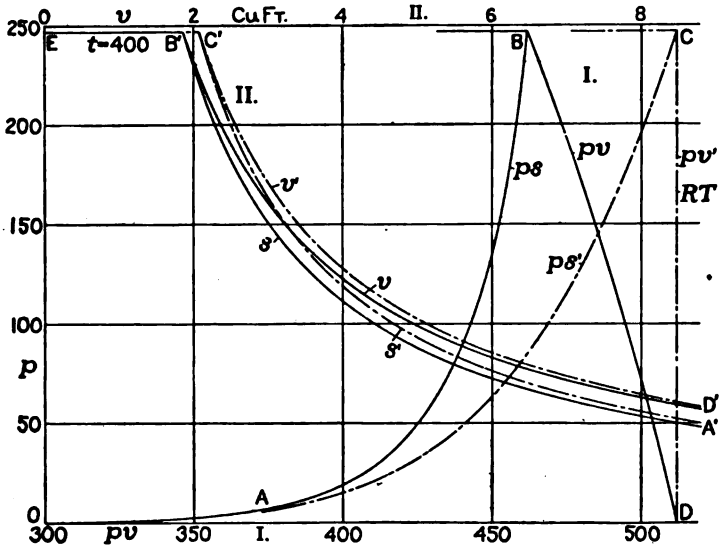

FIG. 44. — Isothermal Curves for Steam, Wet and Superheated. Group I, Plot of pv on p: Group II, Plot of v on p.

In the group of simple volume curves, C'A' shows the ideal saturation volume s', B'A' the actual volume s: curve C'D' is the ideal isothermal $pv' = C$, and B'D' is the actual isothermal. Note particularly that the complete isothermal for steam is made up of two distinct parts: first the constant-pressure isothermal of vaporization EB', then the curve B'D' for superheated steam, the two meeting in a sharp corner at B'.

The three cases of constant-pressure, constant-volume, and constant-temperature change are properly considered here, as among the fundamental properties of steam. Other conditions of expansion, like the adiabatic, will be taken up after the thermal relations have been set forth.

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§ 13. Thermal Properties of Steam

(*a*) SPECIFIC HEAT OF WATER. — As stated in § 11 (*a*), the first step in "making steam" is to raise the water to boiling point. The heat

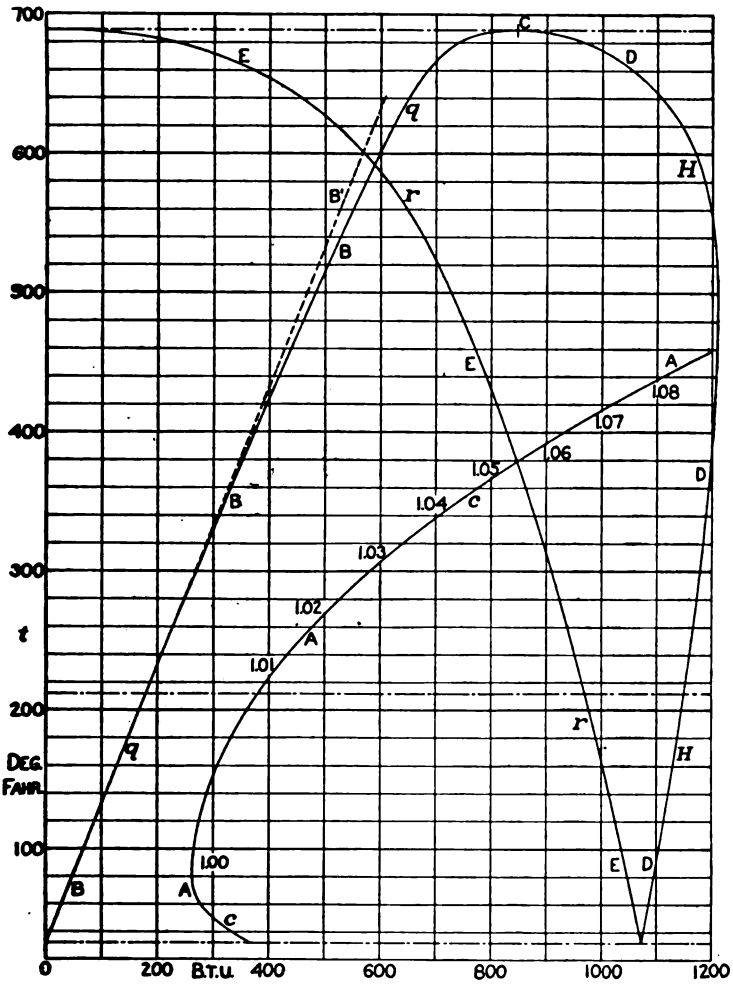

FIG. 45. — Heat Curves for Saturated Steam.

Vertical base, temperature fahrenheit; horizontal ordinates in B.t.u. except for curve A.

Curve A, specific heat *c* of water, given in col. 6 of Table III; scale marked along curve: § 13 (*a*).

Curve BB, heat of the liquid *q*; straight line BB′ shows what *q* would be if *c* were constant at unity: § 13 (*b*), (*e*).

Curve DD, total heat *H*; *q* and *H* meet at the critical point C: § 13 (*c*), (*e*).

Curve EE, latent heat *r*, also shown by the intercept between BB and DD: § 13 (*c*), (*e*).

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required depends upon the specific heat of water, which varies according to the curve marked A on Fig. 45. From about 110 deg. fahr. up to about 530 deg., this curve has an equation of the form

$$c = a - bt + dt^2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (69)$$

a, b, and d being constant coefficients. Below 110 deg. the curve is a plot of experiment, not formulized; above 530 deg., the specific heat c gradually departs from this second-degree equation, in the direction of more rapid increase. The mean value of c from 32 deg. to 212 deg. is 1.00, meeting the definition of the British thermal unit in § 5 (*b*), according to which it takes exactly 180 B.t.u. to raise one pound of water through these 180 degrees. This is called the mean thermal unit; from curve A it appears that the actual or local value of c is unity at about 50 deg. and again at about 153 deg. fahr.

Numerical values of the specific heat of water are given in col. 6 of Table III.

(*b*) HEAT OF THE LIQUID. — In col. 6 of Table II is given the value of q, the heat required to raise one pound of water from 32 deg. to the temperature t in col. 0. It is derived, of course, by means of the relation

$$q = \int c dt. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (70)$$

If the specific heat of water were constant at unity, q would be the same as $(t - 32)$; actually, it differs from this number by an amount which, above 212 deg., increases at a growing rate. At 400 deg., for instance, $(t - 32)$ is 368 and q is 373.9, an excess of 5.9 B.t.u. over the number of degrees of rise.

Although the ice-melting temperature 32 deg. fahr. is naturally and properly taken as the starting point for measurement of heat of the liquid as given in the steam table, it is never the initial temperature for the operation of heating in the boiler. Instead, the feed water enters the boiler at some temperature t_0, and the heat required to raise it to the boiling point or vaporization temperature is $(q - q_0)$.

(*c*) HEAT OF VAPORIZATION AND TOTAL HEAT. — The heat required, at the boiling point t, to turn the pound of water into steam is called the heat of vaporization; its symbol is r, and it is given in col. 7 of Table II. Since this heat all goes into the steam without causing any change of temperature, it is often called latent (hidden) heat.

The sum of q and r, or the heat required for the whole operation of raising one pound of water from 32 deg.to t deg. and there completely evaporating it under constant pressure is called the total heat of saturated steam. The value of

$$H = (q + r) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (71)$$

is given in col. 8 of Table II.

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(*d*) PARTIAL VAPORIZATION AND HEAT OF FORMATION. — If the pound of water is not completely vaporized, the fraction x being changed into steam while $(1 - x)$ or m remains as hot water, only the fraction x of the latent heat is taken up. For the general case of steam formation from water originally at t_0 and with vaporization to the degree x, the heat absorbed per pound of water substance, called the "heat of formation" of steam, is

$$Q = (q - q_0) + xr. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (72)$$

It is often more convenient, as a method of calculation, to follow the formula

$$Q = H - q_0 - mr, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (73)$$

subtracting from the total heat, first the heat q_0 not required because it is already in the feed water, second the portion mr of the latent heat, not required because the m part of the pound of water remains unevaporated.

Using the word "steam" in a general way to cover the steam and water mixture (wet steam) which usually leaves the boiler and is delivered to and works within the engine, we call the fraction x the "quality" of the steam. Steam with 2 per cent of moisture ($m = 0.02$) has a quality of 0.98, or is 98 per cent dry.

EXAMPLE 5. — Steam is made at a pressure of 120 lb. absolute from water at 142 deg., and leaves the boiler with 1.5 per cent of moisture. Find

(*a*) Heat to raise water to boiling point.
(*b*) Heat for vaporization.
(*c*) Heat of formation.

For $p = 120$, $t = 341.3$ (Table I). From Table II, for $t = 341.3$, $q = 312.2$; at 142 deg., $q_0 = 109.8$; then

$$(a) \quad q - q_0 = 312.1 - 109.8 = 202.4 \text{ B.t.u.}$$

The value of r at 341.3 deg. is 877.6; the moisture effect mr is therefore $0.015 \times 877.6 = 13.2$ B.t.u. Thus to compute mr and then subtract it is usually easier than to multiply r by x, and gives here

$$(b) \quad xr = 877.6 - 13.2 = 864.4,$$

as the heat required for the actual amount of vaporization.

Now the total heat requirement is

$$(c) \quad Q = (q - q_0) + xr$$
$$= 202.4 + 864.4 = 1066.8 \text{ B.t.u.}$$

EXAMPLE 6. — In a boiler test the feed temperature was 185.6 deg., the steam pressure 113.6 lb. by gage, the quality of steam 0.982. Find (*a*) the heat that

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would be required to make one pound of dry steam; (*b*) the heat of formation of the actual steam.

$$p = 113.6 + 14.7 = 128.3 \text{ lb. absolute}; \quad t = 346.4 \text{ deg.}$$

$$q_0 = 153.5 \qquad H = 1191.0$$
$$r = 873.5 \qquad m = 0.018 \qquad mr = 15.7$$

Then

(*a*) $Q' = H - q_0 = 1191.0 - 153.5 = 1037.5$ B.t.u.

(*b*) $Q = H - q_0 - mr = 1037.5 - 15.7 = 1021.8$ B.t.u.

(*e*) HEAT CURVES. — On Fig. 45 are plotted the heat quantities just described, heat of the liquid q in curve B, total heat H in curve D These meet, in tangency, at the critical "point" C; at this temperature the liquid state vanishes, merging into the gaseous state without the intermediate stage of vaporization. Latent heat r is included between curves B and D, and is also laid out independently in curve E, from the zero line at the left edge of the diagram. Note that r is a decreasing function of t, and becomes zero at the critical temperature. The total heat increases to a maximum at about 490 deg. (see Table II), then decreases. The straight line B′ shows how q would vary if the specific heat c were constant as 1.00.

(*f*) EXTERNAL WORK AND INTERNAL ENERGY. — During vaporization the volume change u — see § 12 (*d*) — is effected against the pressure p. This involves the performance of external work to the amount Pu or $144pu$ ft. lb., computed as in § 7 (*d*). Reduced to heat units this gives

$$APu = \frac{144}{778} pu = 0.1851pu \text{ B.t.u.} \quad . \quad . \quad . \quad . \quad (74)$$

The external work thus done absorbs a portion of the heat of vaporization r, and the remainder

$$l = r - APu \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (75)$$

is what is expended in the disgregation work — see § 7 (*a*) — of changing the substance from liquid to gas. This quantity l is called the inner latent heat of the steam.

At the evaporation temperature t the pound of water has the volume w; and whatever work is required to bring it to this volume against the pressure p is here supposed to be included in the water heat q, so that

$$k = q - APw \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (76)$$

is the internal work of raising the temperature of the water. Then

$$K = k + l = H - APs \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (77)$$

is the total internal energy of the pound of dry steam.

The external work quantities APw (for the water) and APu (for vaporization) are given in Table III, cols. 7 and 8, while the internal

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energies l and K have a more prominent place in cols. 9 and 10 of Table II. With this tabulation, the internal energy of one pound of steam in the condition defined by quality x or moisture m is to be found, not as

$$I = k + xl, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (78)$$

but rather as

$$I = K - ml. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (79)$$

In the second expression we follow the analogy of Eq. (73); but internal energy I is always to be taken from the zero at melting point, so that there is no occasion for anything like the subtraction of q_0 in Eq. (73).

EXAMPLE 7. — At a certain stage in the process within an engine, steam at the pressure 56 lb. has the quality $x = 0.72$; what is its internal energy?

For $p = 56$ we enter Table II with $t = 288.25$, getting

$$K = 1096.1 \qquad l = 838.8$$

then

$$I = 1091.6 - (0.28 \times 838.8)$$
$$= 1091.6 - 234.9 = 861.2 \text{ B.t.u.}$$

(g) EXTERNAL ENERGY OF THE WATER. — Presupposing an expansion from zero volume under constant pressure, we may call the product Pv the total external energy of the substance. In the case of the water in a boiler (at steam temperature), this energy or work really comes from two sources. Letting w_0 be the volume at feed temperature, Pw_0 is the work done by the feed pump in forcing the water into the boiler; and only the expansion work $P(w - w_0)$ is truly done at the expense of heat supplied. Then in getting from our steam table the value of the heat of formation Q, we ought in strict accuracy to subtract the feed-pump work — or else the table should be so made out that q will be only the heat supplied after the water gets into the boiler.

There are good reasons — from considerations of convenience in making some important thermodynamic calculations, to be explained presently — why it is better to have H or Q include the total external energy APv. The actual value of APw_0 is very small, not reaching one B.t.u. at the upper limit of the usual range of boiler pressures — see Example 8. This discrepancy is far within the probable error of any boiler or engine test, and is quite a little less than the possible inaccuracy of the laboratory experiments upon which are based the values of q and r at high pressures. For these reasons we shall follow Examples 5 and 6 in the not quite theoretically correct practice of taking the tabular q to be the heat supplied within the boiler — when making calculations by Eqs. (71) to (73) for any ordinary conditions — even though in perfect consistency it must be considered as including

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also the work of the feed pump. In the table the latter assumption is followed, in that K is got from H by the method of Eq. (77), or by the subtraction of the full APs. For example, at 400 deg. saturation, from Table III, $APw + APu = APs = 0.86 + 84.65 = 85.51$; from Table II, $H - K = 1201.86 - 1116.35 = 85.51$.

EXAMPLE 8. — If steam is made at a pressure of 300 lb. absolute, how great an error is involved in calculating the heat of formation Q as if the water heat q in the table did not include the work of the feed pump?

The feed water may have a temperature anywhere from 100 deg. to 212 deg., or w_0 may be from 0.0161 to 0.0168 cu. ft. Then APw_0 or $0.185\ pw_0$ will be from 0.89 to 0.92 B.t.u. Subtracting this from the heat to be supplied changes H from 1205.9 to 1205.0, and would change Q by the same amount. Generally, the error involved in using the tabular value directly would be insignificant in anything like a boiler test; but if the physical data for q ever become of sufficient accuracy to give a real meaning to the correction just indicated, it can easily be made.

Comparing this feed-pump work of about 0.90 B.t.u. with $APw = 1.05$ as got by interpolation in Table III, we see that 0.15 B.t.u. is the portion of the "total external energy" of the water which in this case really comes from the heat supplied.

(*h*) SPECIFIC HEAT OF SUPERHEATED STEAM. — In the production of superheated steam (as distinguished from its use), the fundamental operation is heating under constant pressure, from saturation temperature t_s to some higher temperature t. Knowing the value of the specific heat c_p (which is not a constant, as it is for a gas), we can get the heat h_s to be added by the operation

$$h_s = \int_{t_s}^{t} c_p dt. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (80)$$

The data are in such shape that summation by short finite intervals will take the place of true integration. If the mean specific heat over the range from t_s to t is known, an equivalent calculation is

$$h = c_{pm}(t - t_s). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (81)$$

The specific heat under constant pressure is diagrammed in Fig. 46, with c_p as ordinate, on a temperature base. The principal curves, of the type marked PP, show c_p at a particular pressure, varying with the temperature; these curves are, however, designated by the saturation temperature rather than by the pressure, because they are determined by chosen, simple-number values of this initial temperature. In general, c_p is high at the start (against the saturation line SS), drops off rapidly but at a decreasing rate, and after passing a minimum begins to rise slowly. The highest curve drawn, that for 550 deg. saturation or

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for 1045 lb. pressure, barely reaches its minimum within the limits of this diagram. From one pressure to another, c_p rises with the pres-

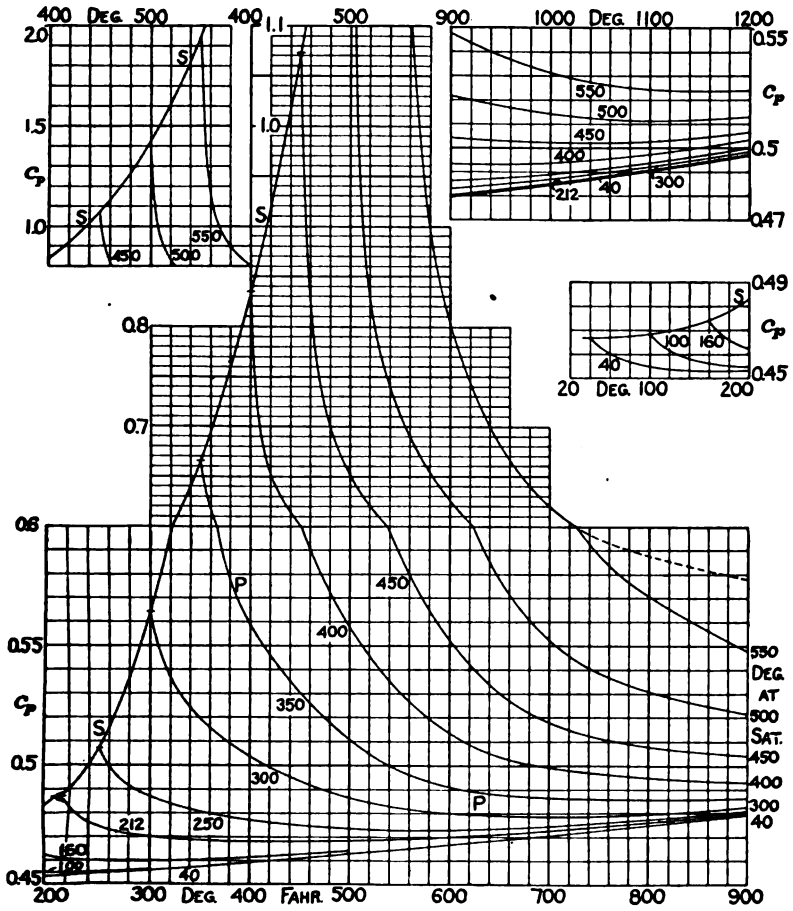

FIG. 46. — The Specific Heat of Superheated Steam under Constant Pressure, c_p.

The small separate sections complete the main diagram at top, right, and left. Scale of c_p is varied as convenient. Curves of the class PP show c_p on t, for the pressure corresponding to the saturation temperature by which each curve is designated. Curve SS is the saturation line, at which superheating begins, and its ordinate is the initial, high value of c_p.

sure: outside of the region near the saturation line, the ratio of c_p to p is, roughly, almost constant.

(*i*) THE TOTAL HEAT OF SUPERHEATED STEAM. — To the total heat H of saturated steam add the superheat h_s; the result,

$$h = H + h_s, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (82)$$

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may properly be called the total heat of superheated steam. It is to be defined as the total amount of heat required to start with one pound of water at 32 deg., and turn it into steam at pressure p and (any) temperature t, the whole operation being carried out under pressure p. The same definition will do for H — see Art. (*c*) — but for saturated steam we may say "steam at pressure p *or* temperature t."

This important heat quantity is very fully laid out in Table VII, for which Fig. 47 serves as an explanatory diagram. When h is thus

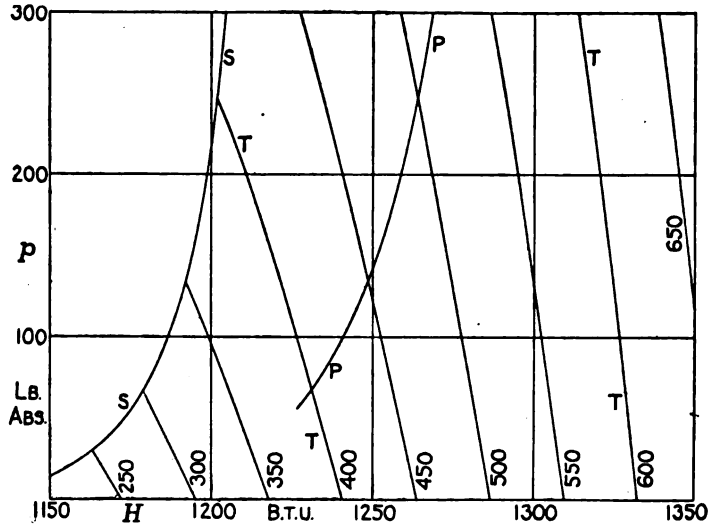

FIG. 47. — The Total Heat of Superheated Steam.

Vertical base, absolute pressure in pounds per square inch; horizontal ordinate, total heat in B.t.u. Curve SS is the saturation line, showing H from Table II, col. 8. Curves of the class TT are lines of equal temperature; PP is a line of equal superheat. This diagram is identical in terms with Table VII. Note that H as marked at the base of the diagram corresponds with the general symbol h in the text.

plotted on p, the important fact appears that the isothermal TT is nearly a straight line. Close to the saturation limit there is a considerable bending (corresponding to the high values of c_p existing in that region as shown by Fig. 46); but away from saturation the isothermal straightens, and as the temperature rises the slant diminishes. This all meets the rational requirements of the situation, for as the steam gets farther away from saturation it should approach a perfect gas in behavior, and for a perfect gas the isothermal will be a vertical straight line in the scheme of Fig. 47.

The curve PP on Fig. 47 represents a class of lines of equal superheat which are drawn on Table-diagram VII and designated by their degrees

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of superheat. They are convenient when the idea of amount of superheat is more prominent than that of actual temperature. In any case, they present at once the connection between the two.

(*j*) INTERNAL ENERGY OF SUPERHEATED STEAM. — Instead of following the analogy of H and h, we let I represent the internal energy of the pound of superheated steam. Very obviously,

$$I = h - APv. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (83)$$

EXAMPLE 9. — For one pound of steam at 150 lb. absolute and at 550 deg., find superheat, total heat, and internal energy.

For 150 lb. the saturation temperature is ..... t_s = 358.5 deg.
The superheat in degrees is therefore ..... $(t - t_s)$ = 191.5 deg.
From Table VII, the total heat h is ..... 1298.5 B.t.u.
From Table II, the total heat H is ..... 1193.8 B.t.u.
The superheat in thermal units is then ..... h_s = 104.7 B.t.u.
For the volume v,
at 358.5 deg. saturation ..... s' = 3.248
$R/p \times (t - t_s) = 0.003970 \times 191.5$ ..... = 0.760
so that ..... v' = 4.008
At 150 lb. and 550 deg., $f_p f_t = 1.214 \times 0.0581$ ..... = 0.071
and the volume is ..... v = 3.937

Now the total external work is

$$APv = 0.185 \times 150 \times 3.937 = 109.3 \text{ B.t.u.}$$

Finally, by Eq. (83), the internal energy is

$$I = 1298.5 - 109.3 = 1189.2 \text{ B.t.u.}$$

(*k*) SPECIFIC HEAT AT CONSTANT VOLUME. — For a perfect gas there are two simple relations between c_p and c_v, namely,

$$c_p - c_v = \text{const.}, \qquad \frac{c_p}{c_v} = \text{const.}$$

With superheated steam, especially when near to saturation, neither of these statements is true; but following the method of § 7 (*b*), as represented by the equation $c_v = c_p - AR$, and finding the value of the quantity which corresponds to AR in Eq. (22), we can readily derive c_v from c_p for steam at any particular condition.

The rate of expansion, as worked out in Example 3, § 12 (*j*), is the volume increment of one pound of steam, in cubic feet per degree of temperature rise, under the pressure p. The accompanying rate of external work-performance, analogous to AR in Eqs. (21) and (22), is very evidently

$$A\frac{dU}{dt} = AP\left(\frac{dv}{dt}\right)_p . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (84)$$

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In the example just referred to, the pressure was 150 lb., and the rate of expansion was found to be 0.00521 cu. ft. per degree at saturation and 0.00460 cu. ft. at 500 deg. Using $0.185p$ for AP as heretofore, we multiply these values by $0.185 \times 150 = 27.75$, and get 0.145 and 0.128 for the respective rates of external work, expressed in B.t.u. instead of foot pounds — compare the constant $AR = 0.0685$ for air, in § 7 (*b*). Interpolating in Fig. 46 for $t_s = 358.5$ deg., we find that c_p is about 0.692 at saturation and about 0.494 at 500 deg. Then by subtraction,

$$\text{at saturation, } c_v = 0.692 - 0.145 = 0.547;$$
$$\text{at 500 deg., } c_v = 0.494 - 0.128 = 0.366.$$

Note that these two values of c_v are not on the same volume line; they give conditions on two volume lines, at the points where these cross the pressure line for 150 lb.

There is no occasion for a general working out of c_v and the drawing of a diagram like Fig. 46. To get the heat added in a heating at constant volume, all that need be done is to calculate the internal energy at the initial and at the final state and subtract the first from the second.

EXAMPLE 10. — If one pound of steam is confined in a space of 4 cu. ft. and raised from 350 deg. to 700 deg., how much heat is imparted to it?

By interpolation on the curve for 4 cu. ft. in Table VI, the pressures are found to be,

$$\text{at 350 deg., } p_1 = 114.0 \text{ lb};$$
$$\text{at 700 deg., } p_2 = 171.5 \text{ lb}.$$

Referring to Table VII, the total heats are,

$$\text{at 114.0 lb. and 350 deg., } h_1 = 1202.4 \text{ B.t.u.};$$
$$\text{at 171.5 lb. and 700 deg., } h_2 = 1371.4 \text{ B.t.u.}$$

Calculation of the external work as $0.185\ pv$ gives,

$$\text{at 350 deg., } AU_1 = 0.185 \times 114.0 \times 4 = 84.3 \text{ B.t.u.};$$
$$\text{at 700 deg., } AU_2 = 0.185 \times 171.5 \times 4 = 126.8 \text{ B.t.u.}$$

Subtraction of this AU or APv from h gives I;

$$\text{at 350 deg., } I_1 = 1202.4 - 84.3 = 1118.1 \text{ B.t.u.};$$
$$\text{at 700 deg., } I_2 = 1371.4 - 126.8 = 1244.6 \text{ B.t.u.}$$

The difference, $I_2 - I_1 = 126.5$ B.t.u., is the quantity sought.

The mean specific heat, over the 350 deg. of rise, is

$$c_{vm} = 126.5 \div 350 = 0.362.$$

The same general method is equally applicable when the operation lies wholly or partly within the region of wet steam, or of partial condensation.

(*l*) ENTROPY OF STEAM. — Entropy is a very useful quantity in thermodynamic calculations for steam. Corresponding to the heat quantities q, r, and H, we have the entropy values given in cols. 11, 12,

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and 13 of Table II. To get more convenient symbols, a is used for N_q, the entropy of the liquid, and b for N_r, the entropy of vaporization; then N is the total entropy, the sum of a and b. The methods of calculating a and b are,

$$a = N_q = \int \frac{dq}{T} = \int_{491.6}^{T} \frac{c\,dt}{T}; \quad \ldots\ldots \quad (85)$$

and, since r is imparted isothermally,

$$b = N_r = \frac{r}{T}. \quad \ldots\ldots \quad (86)$$

In being superheated at constant pressure, the entropy acquired is

$$n = \int_{T_s}^{T} \frac{c_p dt}{T} \quad \text{or} \quad \int_{T_s}^{T} \frac{c_p}{T} dt. \quad \ldots\ldots \quad (87)$$

The second way of writing the expression indicates the useful method of computing the ratio c_p/T and then integrating or summing this on a temperature base. Also, with short enough intervals, say of ten degrees, there is no appreciable error in the simple summing of $\Delta n = \Delta h/T_m$; that is, Δn is got by dividing the heat added by the mean absolute temperature during the interval.

(*m*) THE ENTROPY DIAGRAM. — The fundamental temperature-entropy curves are given in Fig. 48. For any pressure, AB represents the reception of the water heat q and BC the reception of the latent heat r, while CD shows superheating at constant pressure — the area under each line being, of course, equal to the heat transferred. In terms of the tabular quantities, length QB is a or N_q, and length BC is b or N_r, so that QC is N. For superheat conditions, n_s is added to this N, to get n as diagrammed in Table VIII. The constant-pressure curves here shown, with starting points at from 40 deg. to 550 deg. on the saturation line, were computed as the basis for that table, and the intermediate curves were located by interpolation. Keep clearly in mind that these lines are determined by particular values of pressure p, even though they are marked with the corresponding values of temperature on the saturation line.

The origin of entropy, at the left edge of Fig. 48, is determined by the thermal condition of water at 32 deg. fahr., the same zero being taken, of course, as for heat measurement. The curve ABK is not only a plot of entropy a, but is also a curve of operation; the saturation line SCK is, however, to be thought of as a locus of condition or line of relation, not generally representing an operation. The critical point at K corresponds to C in Fig. 45. On this thermal diagram, K does not appear to be so very far above the upper limit of our service tables, at 550 deg. saturation; but in terms of pressure the distance is relatively much greater.

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Of the three fundamental simple operations, heating under constant pressure is completely represented by the broken line AB - BC - CD,

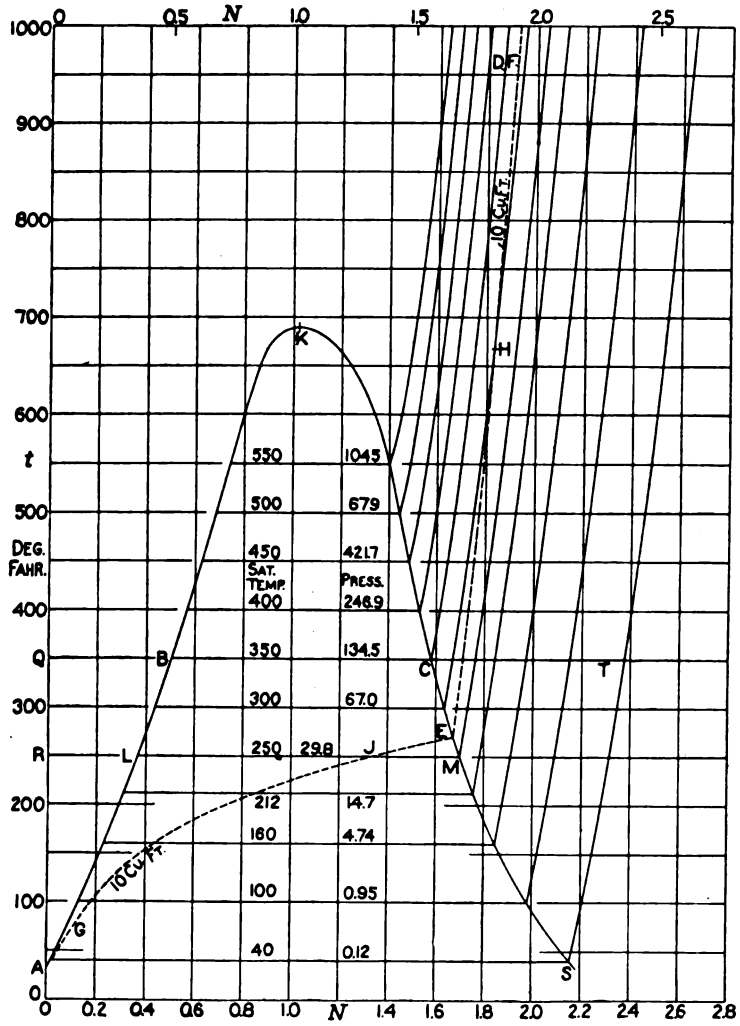

FIG. 48. — The Temperature-entropy Diagram for Steam.

Vertical base, temperature fahrenheit; horizontal ordinate, entropy above state of water at 32 deg. Line ABK shows operation of water heating, up to any point B; BC is the operation of vaporization, CD that of superheating at constant pressure. The saturation line SCK serves chiefly as a locus of C or as a boundary between wet and superheated steam. Curve GEF is a line of constant-volume heating.

very distinctly divided into its several parts. The isothermal is identical with BC, and extends into the region of superheat as CT. One

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line of constant volume is fully given in GE - EF: above E, points on this curve are found by combining readings from Tables VI and VIII, not by using c_v in an expression analogous to Eq. (87). Inside of the saturation line, the volume can be kept constant with falling pressure only by a partial (and rapidly increasing) degree of condensation.

EXAMPLE 11. — Find points on the constant-volume line GEF in Fig. 48. As indicated, this line is drawn for $v = 10$ cu. ft.

First, get the point J, on the evaporation line at 250 deg. The saturation volume is $s = 13.81$ cu. ft., so that for 10 cu. ft. the quality must be $x = 10 \div 13.81 = 0.724$. Entropy b has the value 1.331, and in evaporation to $x = 0.724$, LJ $= 1.331 \times 0.724 = 1.109$ is added to $a = 0.382$ at L, giving 1.491 as the complete ordinate RJ.

The point H, where the line for $v = 10$ cu. ft. crosses the constant-pressure line from 300 deg. saturation, is readily found by referring to Table VI and noting that this intersection there comes at about 668 deg. When seeking a point on a constant-pressure line not drawn on Fig. 48, as that at 290 deg. saturation, for instance, we first look in Table VI, and find that $t = 514$ deg. for this pressure and 10 cu. ft.; and from the 290-deg. line in Table VIII we get $n = 1.778$ for $t = 514$, and thus have the two coördinates of a point on the curve EF, Fig. 48.

§ 14. Various Curves and Operations

(*a*) THE THREE PRIMARY OPERATIONS, those at constant pressure, at constant volume, and at constant temperature, have been fully discussed in § 12 (*j*) to (*n*) and in § 13 (*m*). Adiabatic expansion, the curve of constant quality, and the equilateral hyperbola $pv = C$ are considered in the section which follows. The line of constant total heat, at first sight a simple curve of relation, is really so closely connected with the steam-jet cycle that the next chapter seems to be the more appropriate place for its presentation, in § 17 (*b*).

(*b*) ADIABATIC EXPANSION. — As has been pointed out in § 9 (*c*), an adiabatic operation takes place at constant entropy, or is isentropic: that there can be no change of entropy without transfer of heat has been further emphasized in § 10 (*d*). The vertical lines of constant entropy in Fig. 48 (inclined in Table VIII) are therefore lines of adiabatic operation, expansion or compression. Inside of the saturation limit — anywhere within the region ABKCS, Fig. 48, including points on the boundary lines — relations are determined by computation: in the superheat region, only graphical methods are really available.

For a steam and water mixture, as at J, Fig. 48, the entropy is, in analogy to Eq. (72),

$$n = a + xb = \text{RL} + \text{LJ}; \quad . \quad . \quad . \quad . \quad . \quad . \quad (88)$$

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or, following the idea of Eq. (73),

$$n = N - mb = \text{RM} - \text{JM}. \quad . \quad . \quad . \quad . \quad . \quad (89)$$

Here n is used as a general symbol for total entropy of wet steam, and N for the particular, limiting condition of dry saturation, as in Table II.

To establish the adiabatic relation between two states designated as 1 and 2, say at H and K on Fig. 49, it is necessary to satisfy the condition

$$a_1 + x_1 b_1 = a_2 + x_2 b_2, \quad . \quad . \quad . \quad . \quad . \quad . \quad (90)$$

Most frequently, x_2 is the unknown quantity, to be got, of course, by the calculation

$$x_2 = \frac{(a_1 + x_1 b_1) - a_2}{b_2} = \frac{\text{AH} - \text{DE}}{\text{EG}}. \quad . \quad . \quad . \quad . \quad (91)$$

The alternate method, rather preferable for slide-rule work because the numbers to be handled are relatively smaller in most cases, is to get the moisture fraction $m_2 = (1 - x_2)$, by the relation

$$m_2 = \frac{\text{KG}}{\text{EG}} = \frac{N_2 - n_1}{b_2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (92)$$

In § 7 (h) was stated the general principle that in adiabatic expansion the external work done just equals the loss of internal energy. Letting 1 designate the initial (high-pressure) state and 2 the final (low-pressure) state, and using Eq. (79) for the internal energy I, we have

$$AU = I_1 - I_2 = (K_1 - m_1 l_1) - (K_2 - m_2 l_2). \quad . \quad . \quad (93)$$

EXAMPLE 12. — Let a space of 4 cu. ft. be filled with steam 350 deg. or 134.5 lb. which contains 2 per cent of moisture; if it be expanded adiabatically to 212 deg. or 14.7 lb., what will be the final condition and the amount of external work done?

The tabular quantities needed are,

At 350 sat.	At 212 sat.
$s_1 = 3.345$	$s_2 = 26.78$
$a_1 = 0.5032$	$a_2 = 0.3120$
$b_1 = 1.0753$	$b_2 = 1.4439$
$N_1 = 1.5785$	$N_2 = 1.7559$
$l_1 = 787.7$	$l_2 = 896.9$
$K_1 = 1108.6$	$K_2 = 1076.9$

The initial specific volume is 2 per cent less than s_1, or $v_1 = 3.345 - 0.067 = 3.278$ cu. ft.: the weight of steam involved is therefore $4 \div 3.278 = 1.2203$ lb.

The initial total entropy per pound is,

$$n_1 = 1.5785 - (0.02 \times 1.0753)$$
$$= 1.5785 - 0.0215 = 1.5570.$$

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Now applying Eq. (92),

$$m_2 = \frac{1.7559 - 1.5570}{1.4439} = \frac{0.1989}{1.4439} = 0.13774.$$

The dry-steam volume 26.78 is therefore diminished by 13.77 per cent of itself, or by 3.68, to 23.10 cu. ft. For the whole weight of steam present, the final volume is $23.10 \times 1.2203 = 28.19$ cu. ft.

For I_1 we have $1108.6 - (0.02 \times 787.7) = 1108.6 - 15.8 = 1092.8$; for I_2, $1076.9 - (0.1377 \times 896.9) = 1076.9 - 123.5 = 953.4$; then the external work is $1092.8 - 953.4 = 139.4$ B.t.u. per pound of steam, or $139.4 \times 1.2203 = 170.1$ B.t.u. total.

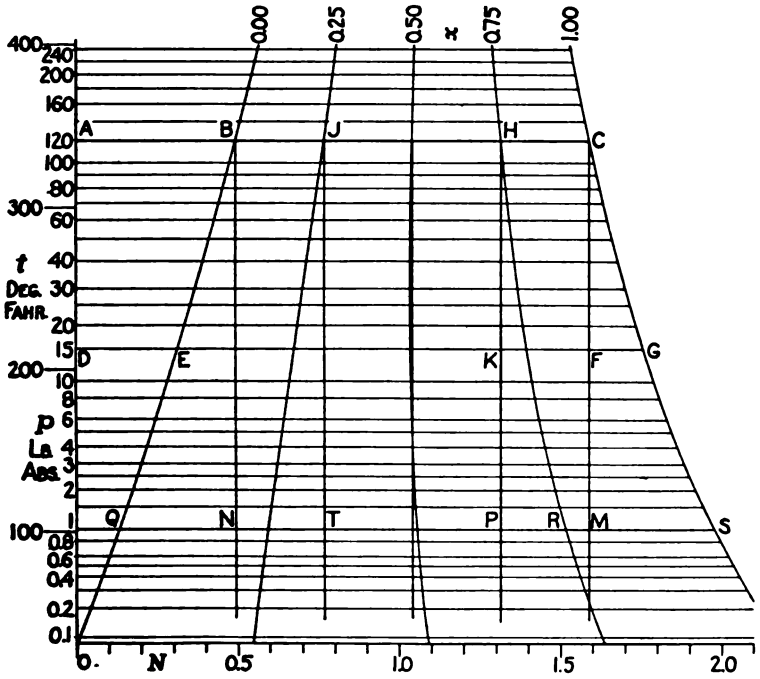

FIG. 49. — Change of Quality in Adiabatic Expansion.

Same coördinates as in Fig. 48. Lines of class HP are adiabatics or isentropics; those of class HR are curves of constant quality or of constant steam weight.

EXAMPLE 13. — If one pound of steam at 150 lb. absolute (358.5 deg. sat.) and at 500 deg. fahr. is expanded adiabatically to 60 lb. (292.7 deg. sat.), what is its final temperature and what its initial and final volumes?

The initial volume has been found in Example 2, page 74, to be 3.714 cu. ft.

In Table VIII the initial entropy is read as 1.660; this would cross a line for 292.7 deg. sat. at about 316 deg., which is therefore the final temperature sought.

From Table VI the corresponding volume is found to be about 7.43 cu. ft.

(*c*) THE CHANGE IN QUALITY or condition during adiabatic operation is best illustrated by the method of Fig. 49, which consists in drawing,

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on the TN diagram, curves of constant quality like HR. Each of these is for a certain value of x, or it divides all the b or N_r lengths, such as BC and QS, in a constant ratio. If adiabatic expansion begins with dry steam as at C or with steam weight predominant as at H, there is condensation as the pressure drops, shown by the swing of the x-constant lines CS and HR from the adiabatic lines CM and HP toward the right: steam is condensed to supply heat for external work. Starting with hot water as at B or with predominant water weight as at J, there is evaporation during expansion: the lowering of temperature releases an increasing proportion of the water heat q, and what is not needed for external work evaporates water. With half steam and half

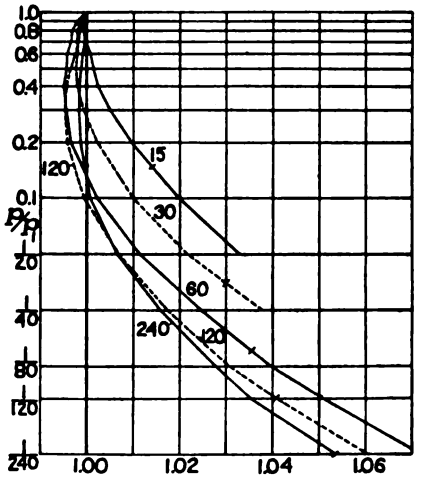

FIG. 50. — Comparison of Adiabatic Curves.

Vertical base, ratio p/p_1, to a scale made uniform for the logarithm of this ratio; horizontal ordinate, relation between values of the pv ratios in Table 5.

water at the start, the two tendencies (toward condensation and toward evaporation) are just about equalized, as is shown by the nearly vertical course of the curve for $x = 0.5$.

(*d*) FORM OF THE ADIABATIC CURVE. — It would greatly simplify some important calculations, to be described in the next chapter, if the adiabatic pressure-volume curve could be represented by an equation of the form $pv^n = C$. The exact form of this curve can be found only by means of the method illustrated in Example 12. In Table 5 and in Fig. 50 are given the results of an investigation into this matter.

Five curves are computed, for initial pressures p_1 of the values 240, 120, 60, 30, and 15 lb. absolute; the initial quality x_1 is unity, or the steam is dry-saturated at the start. The pressure ratio p/p_1, used

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as argument in the table and marked at the left edge of the diagram, is the independent variable. The five columns with pressure headings give the results of the main calculation: the last column is from Zeuner's equation for the adiabatic of saturated and wet steam,

$$pv^n = C, \qquad n = 1.035 + 0.1\,x_1. \quad . \quad . \quad . \quad . \quad (94)$$

For initial dry steam, $n = 1.135$.

TABLE 5. COMPARISON OF ADIABATIC CURVES. RELATIVE VALUES OF THE PRODUCT pv.

$\frac{p}{p_1}$	p_1 240	120	60	30	15	Formula.
0.9	.9866	.9868	.9856	.9862	.9872	.9877
0.8	.9734	.9723	.9708	.9719	.9739	.9738
0.7	.9580	.9566	.9552	.9564	.9582	.9583
0.6	.9402	.9387	.9371	.9392	.9418	.9410
0.5	.9195	.9175	.9167	.9185	.9220	.9206
0.4	.8954	.8928	.8924	.8950	.8990	.8968
0.3	.8650	.8627	.8628	.8660	.8708	.8666
0.2	.8247	.8223	.8230	.8276	.8336	.8258
0.1	.7610	.7600	.7621	.7681	.7755	.7604
$\frac{1}{20}$	.7048	.7050	.7083	.7153	.7233	.7002
$\frac{1}{40}$	.6545	.6555	.6605	.6686	.....	.6443
$\frac{1}{60}$	.6102	.6120	.6174	.....	.....	.5938
$\frac{1}{120}$	.5856	.5890	.5945	.....	.....	.5658
$\frac{1}{240}$	.5493	.5524	.5585	.....	.....	.5209

The quantity given in the table is the ratio of the changing product pv to the initial p_1v_1; in each column we see how this product decreases as the steam expands. The relative trend of the curves can best be seen in Fig. 50, where the true adiabatics are compared with the curve $pv^{1.135} = C$. The vertical line at 1.00 represents this standard, and the broken curves show the departures of the true adiabatics. For example, at $p/p_1 = 0.1$ and in the 60-lb. column, $pv = 0.7621$, in terms of initial p_1v_1 as unity; in the formula column the value is 0.7604. The difference, $0.7621 - 0.7064 = +0.0017$, is not laid off directly, but is first reduced to a fraction or percentage of the formula value; so that $0.0017 \div 0.7604 = 0.0022$ is the distance from the reference line to the point on the curve.

As in a number of diagrams where pressure as base has been plotted on a uniform scale of saturation temperature, it is here desirable to crowd together the high-pressure values and spread out those at low pressures. This effect is obtained in simple fashion by laying off $\log(p/p_1)$ on a uniform scale, which gives the vertical spacing in Fig. 50.

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The diagram makes it evident that the equation does not very effectively represent the true law, especially if the expansion be carried to low pressures. Writing it in the form

$$pv = p_1 v_1 \left(\frac{v_1}{v}\right)^n \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (95)$$

shows that as n is larger, pv is less; conversely, a rapid decrease in pv calls for a high value of n, a slow decrease for a lower value The shape of the true curves in Fig. 50 indicates that n ought to grow less as the pressure falls: at first the true pv decreases a little more rapidly than that by formula, but after passing equality (parallelism on the diagram) it shows a continually less rapid rate of decrease. In Eq. (94), n is supposed to be fixed by initial x_1. If it were made a function of x or of p, down the curve, the equation might be accommodated to the true law, but it would then be too complicated for convenient use. Except for approximations of comparatively short range, the simple formula is not reliable.

That the true adiabatics are of very much the same form, and in practical agreement over a considerable range of initial pressure, is a fact of much significance, which will be used later — see § 16 (*d*) and (*g*).

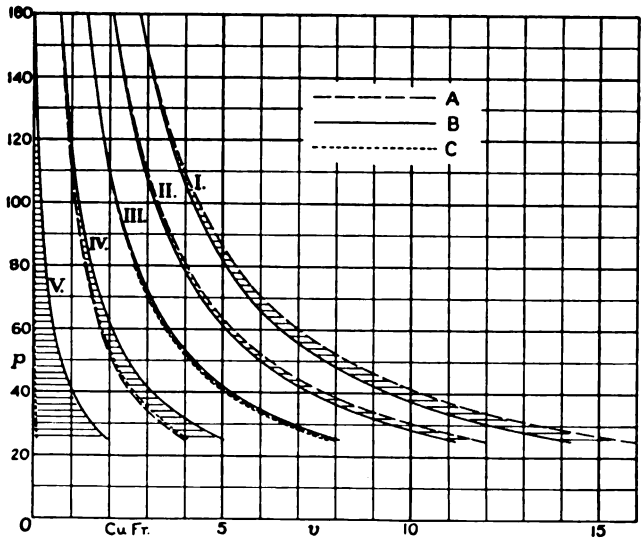

FIG. 51. — Adiabatic Curves for Wet Steam.

A. Constant quality. B. Exact adiabatic. C. Adiabatic by Eq. (94).

(*e*) THE EFFECT OF INITIAL CONDITION upon the form of the adiabatic pressure-volume curve is shown by Fig. 51, which transfers to this plane of representation the indications of Fig. 49. The five sets

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of curves start at 160 lb. absolute, with x_1 equal respectively to 1.0, 0.75, 0.5, 0.25, and 0.0; from each initial point is drawn the curve of constant quality, the true adiabatic, and (where distinguishable) the adiabatic by Eq. (94). The horizontal shading, emphasizing the departure of the adiabatics from constant quality, shows the condensation or evaporation that takes place.

One thing made clearly evident by this diagram is, that as the initial proportion of water is greater, the ratio of expansion, or of v to the v_1 at the start, is greater. The constant-quality lines are essentially alike in respect to this ratio. With high values of x_1, the adiabatic falls below the x-constant line; but as x_1 grows smaller the adiabatic first rises to, then comes well above, the curve of constant quality. With hot water at the start, curve V, the relative increase of volume is very much greater than in the case of curve I.

The approximate Zeuner formula holds fairly well down to $x_1 = 0.7$, but in the lower ranges of x_1 it entirely (and naturally) fails to represent the rapid increase in relative expansion just noted.

The adiabatic of superheated steam conforms approximately to the equation $pv^n = C$, with a value of n in the neighborhood of 1.3; but instead of attempting to apply this equation, with the necessary variation in n, it is better to follow the method of Example 13, correlating temperature and pressure by the entropy relation, then going into Table VI or computing v from data in Table II.

(*f*) THE CURVE OF CONSTANT STEAM WEIGHT. — On the pressure-volume diagram, the curve of constant quality, drawn as reference line in Fig. 51 and corresponding to HR in Fig. 49, is often called the curve of constant steam weight. If a mixture of steam and water expands in a cylinder without condensation or evaporation, the quality x remaining unchanged, there is present a constant weight of steam (and of water). The saturation line B1 - B2 - B3 on Fig. 40 is one particular case of this curve, for $x = 1$; the water-volume line B4 is another, for $x = 0$; and between these lie all possible values of x. The constant-weight curve is useful as a standard of comparison for the curves of the indicator diagram, measuring the condensation or evaporation during expansion and compression; further information concerning this matter will be found in §§ 22 and 23, and in § 27.

(*g*) THE EQUILATERAL HYPERBOLA, $pv = C$, is much used in the analysis of indicator diagrams from the steam engine. The curve has no fundamental place in the theory of the engine, but it serves a useful purpose for two reasons. One is that the hyperbola is easier to lay out than the more logical curve of constant quality — see § 6 (*g*) and Fig. 29: the other reason is, that as the result of the complex thermal reactions

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within the cylinder the observed expansion curve generally approximates quite closely to the form $pv = C$, so that any marked departure from the hyperbola indicates something abnormal.

The plot of pv on p in Fig. 52 is intended to illustrate and make quantitative the change in quality involved in expansion along the hyperbola. Any vertical line on the diagram represents the relation $pv = C$, and we need only compare these verticals with the XX curves of constant quality. Suppose, for instance, that in an engine the quality of the steam at cut-off is $x_1 = 0.74$, the pressure being 100 lb. absolute;

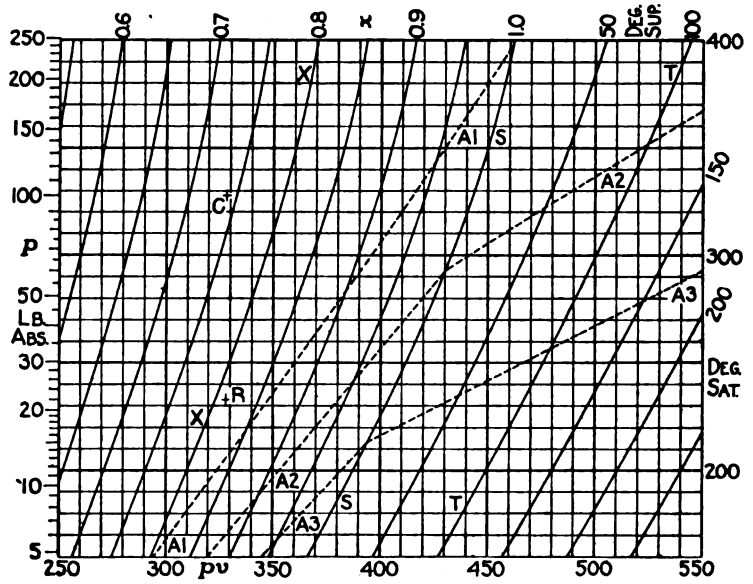

FIG. 52. — Diagram for the Equilateral Hyperbola.

Vertical base, pressure on a scale uniform for saturation temperature; horizontal ordinate (along lines of constant pressure), value of the product pv.

Vertical lines represent $pv = C$.

Curves of class SS, XX, TT, lines of constant quality or of equal superheat, SS being the saturation line.

Curves marked A, adiabatics.

if the expansion curve conforms to $pv = C$, what will be the quality at release, this occurring at 25 lb. absolute?

The initial state is first located, in the point C. Vertically below this, at 25 lb. pressure, is the point R. Reading its location between the XX lines for 0.8 and 0.85, we get $x_2 = 0.814$. The evaporation of moisture amounts, therefore, to 7.4 per cent of the total weight of steam and water in the cylinder.

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In Fig. 52 are plotted also three of the adiabatics from Table 5, those with their saturation points at 240, 60, and 15 lb., in the curves A1, A2, and A3; and the last two are also extended into the region of superheat. It is of interest to note how near these come to being straight lines in the terms of this diagram, on what is really a base of saturation temperature: but in the absence of any workable relation between p and t, there does not appear to be any useful outcome in the way of a direct, simple relation between pv and p.

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CHAPTER IV

IDEAL STEAM CYCLES

§ 15. The Static Pressure Cycle

(*a*) THE CARNOT CYCLE WITH STEAM. — With the physical conditions described in § 8 (*d*) and with the same scheme of working, the Carnot cycle takes the form shown in Figs. 53 and 54 when steam is used as the medium. Starting at 1, there is in the cylinder one pound of water

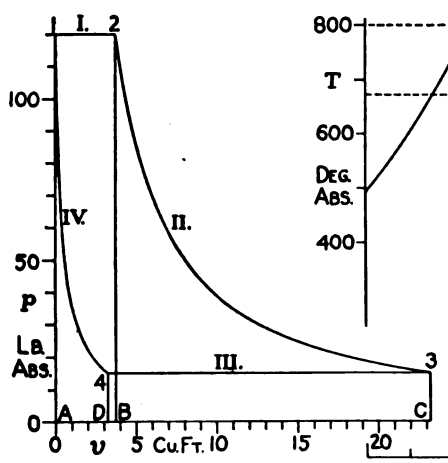

FIG. 53. — Pressure-volume Diagram for the Carnot Cycle.

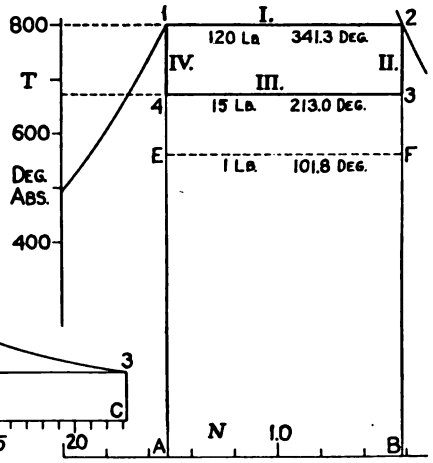

FIG. 54. — Temperature-entropy Diagram for the Carnot Cycle.

These two diagrams are drawn for one pound of steam, and for the same governing conditions; they give parallel illustration of the computations in Example 14.

at temperature t_1 and pressure p_1; and isothermal expansion under constant pressure, to complete evaporation at 2, constitutes the first phase. Then follows adiabatic expansion to the lower limit of pressure and temperature at 3, completing the outstroke. On the return stroke there is first the isothermal compression 34, with condensation and rapid rejection of heat, then adiabatic compression along 41 to the initial state at 1.

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As stated in § 8 (*f*), the efficiency of this cycle is independent of the particular properties of the medium, provided only that the latter is capable of isothermal and adiabatic operations. Its value for steam as for a perfect gas, is therefore,

$$E = \frac{T_1 - T_2}{T_1} = \frac{t_1 - t_2}{t_1 + 459.6}, \quad \ldots\ldots \quad (96)$$

the second expression indicating the more convenient method of calculation. Since the heat received during phase I is r_1, the useful work of the cycle, per pound of steam and expressed in heat units is

$$AU = Er_1. \quad \ldots\ldots\ldots \quad (97)$$

The idea of this cycle will become more concrete, and the statement just made as to efficiency more convincing, by the working out of numerical values for a particular case.

EXAMPLE 14. — With the limiting conditions marked on Fig. 54, determine volume and heat values for the four points of the cycle, and work values for each phase and for the whole cycle.

From Table II, by interpolation,

At p = 100 lb.,		At p = 15 lb.,	
t = 341.31	l = 795.17	t = 213.03	l = 896.15
s = 3.727	K = 1107.00	s = 26.27	K = 1077.14
q = 312.22	a = 0.49197	q = 181.04	a = 0.31353
r = 877.56	b = 1.09573	r = 969.04	b = 1.44068
H = 1189.79	N = 1.58770	H = 1150.08	N = 1.75421

At point 1, for $x_1 = 0.00$,

volume $v_1 = w_1 = 0.018$ cu. ft.;
total heat $h_1 = q_1 = 312.22$ B.t.u.;
inner heat $I_1 = K_1 - l_1 = 311.83$ B.t.u.

At point 2, for $x_2 = 1.00$,

$v_2 = s_2 = 3.727$ cu. ft.;
$h_2 = H_2 = 1189.79$ B.t.u.;
$I_2 = K_2 = 1107.00$ B.t.u.

At point 3,

$$m_3 = \frac{N_3 - N_2}{b_3} = \frac{1.75421 - 1.58770}{1.44068} = \frac{0.16651}{1.44068} = 0.11557;$$

$$v_3 = s_3 - m_3 s_3 = 26.27 - 3.04 = 23.23 \text{ cu. ft.};$$
$$h_3 = H_3 - m_3 r_3 = 1150.08 - 0.1156 \times 969.0$$
$$= 1150.08 - 111.99 = 1038.09 \text{ B.t.u.};$$
$$I_3 = K_3 - m_3 l_3 = 1077.14 - 0.1156 \times 896.2$$
$$= 1077.14 - 103.59 = 973.55 \text{ B.t.u.}$$

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At point 4,

$$x_4 = \frac{a_1 - a_4}{b_4} = \frac{0.49197 - 0.31352}{1.44068} = \frac{0.17845}{1.44068} = 0.12386;$$

$$v_4 = w_4 + x_4 u_4 = 0.017 + 0.1239 \times 26.26 = 3.27 \text{ cu. ft.};$$

$$h_4 = q_4 + x_4 r_4 = 181.04 + 120.02 = 301.06 \text{ B.t.u.};$$

$$I_4 = (K_4 - l_4) + x_4 l_4 = 180.99 + 111.00 = 291.99 \text{ B.t.u.}$$

In phase I,

heat imparted $Q_1 = r_1 = 877.56$ B.t.u.;

external work $AU_1 = AP_1 u_1 = r_1 - l_1$

$= 877.56 - 795.17 = 82.39$ B.t.u.

In phase II,

$$AU_{II} = I_2 - I_3 = 1107.00 - 973.55 = 133.45 \text{ B.t.u.}$$

In phase III, the length 34 represents $(1 - m_3 - x_4) = 0.76057$ of the total change from water to steam; then

$$Q_{III} = 0.76057 \times r_3 = 0.76057 \times 969.04 = 737.03 \text{ B.t.u.};$$

$$AU_{III} = 0.76057 \times AP_3 u_3 = 0.76057 (r_3 - l_3)$$

$0.76057 \times 72.89 = 55.44$ B.t.u.

In phase IV,

$$AU_{IV} = I_1 - I_4 = 311.83 - 291.99 = 19.84 \text{ B.t.u.}$$

Now in terms of the phases, the total work in the outstroke is $A(U_I + U_{II}) = 82.39 + 133.45 = 215.84$ B.t.u.; the negative work of the return stroke is $A(U_{III} + U_{IV}) = 55.44 + 19.84 = 75.28$ B.t.u.; and the net work is $215.84 - 75.28 = 140.56$ B.t.u.

In terms of heat reception and rejection, the heat converted is $Q_I - Q_{III} = 877.56 - 737.02 = 140.54$ B.t.u., and the efficiency is $140.54 \div 877.56 = 0.16015$.

The initial absolute temperature is $T_1 = 341.3 + 459.6 = 800.9$ deg., and the range of temperature $341.31 - 213.03 = 128.28$; then the efficiency according to Eq. (96) is

$$E = \frac{128.28}{800.9} = 0.16017.$$

To eliminate the small discrepancies which appear in the last figure of the results, it would be necessary to have tabular numbers carried to one more place — a degree of apparent accuracy entirely without physical foundation or meaning. As it is, the calculations above are carried one figure beyond what is of any practical significance.

(*b*) THE AVAILABILITY OF THIS CYCLE, or its effective value for adaptation to actual conditions, is far higher with steam than with air as medium. In the example just computed, with a total piston displacement of $V = v_3 - v_1 = 23.21$ cu. ft. (so that mean pressures are obtained through the division of foot-pound numbers by $144 \times 23.21 = 3342$), the work and pressure values which correspond to those computed in § 8 (*h*) are as follows:

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	Work	Mean Pressure
Forward stroke.....	215.84 B.t.u. = 167,920 ft. lb.	50.23 lb. per sq. in.
Return stroke.....	75.28 B.t.u. = 58,570 ft. lb.	17.52 lb. per sq. in.
Net or effective.....	140.56 B.t.u. = 109,350 ft. lb.	32.71 lb. per sq. in.

Here the negative work drops into the position of relative insignificance which was occupied by the effective work in the previous case. This showing justifies the remarks made in § 8 (*i*); and we shall now proceed to consider the principal modifications which unavoidably accompany the actual embodiment of the idea of the Carnot cycle.

Incidentally, it may be remarked that Fig. 54, besides giving a striking illustration of thermal efficiency in the ratio of area 1234 to area 12BA, suggests quite clearly the limits of possible steam-engine performance. The full-line rectangle represents the noncondensing engine, with moderately high boiler pressure. Lowering the exhaust pressure from 15 lb. to 1 lb. absolute — which can be done with a good condenser outfit, although 2 lb. is more usual — will raise the Carnot efficiency from 16 per cent to 30 per cent. If at the same time the initial temperature (of saturated steam) be raised to 400 deg., which is seldom exceeded in practice, the efficiency will be about $300 \div 860 = 35$ per cent. A full consideration of engine performance forms the subject of Chapter VI.

Note that in Fig. 54 the whole length of the absolute-temperature ordinates is shown. Generally, as in Figs. 48, 56, 58, etc., it is better to give only the useful range of temperature, to a much larger scale.

(*c*) The Separation of Function. — In the wholly ideal Carnot engine, the entire cycle — namely, heat reception and expansion, heat rejection and compression — is supposed to be performed within the cylinder, using a confined and unchanged body of the working medium. While not absolutely impossible, this manner of operation is altogether impracticable, from considerations of convenience and of economy. The first step in the adaptation to actual conditions and materials is the separation of function, heat reception being assigned to the boiler, work performance to the engine, heat rejection to the condenser. We have to consider, then, the action of the whole steam plant, of which the engine proper is but one element.

In the division of labor among the principal organs of the plant, phase IV of the Carnot cycle disappears. To realize it, we should have to withdraw from the condenser a proper mixture of water and steam, and let the feed pump compress it, in an approximately adiabatic fashion, to the pressure in the boiler. The possibility of such a process is very incomplete, and its attempted introduction would most undesirably increase the size, the power absorption, and the delicacy in

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action of the boiler-feeding apparatus. In the prevailing type of steam plant, the exhaust is all condensed (whether in a condenser or in the atmosphere), and the resulting water, or more commonly an equivalent amount of new water, is pumped into the boiler.

The ideal cycle which really underlies the action of the steam plant is therefore not the Carnot cycle, but is what is generally called the Rankine cycle or the Clausius cycle.

(*d*) THE RANKINE CYCLE. — The diagrams in Figs. 55 and 56 represent the following cycle of operations:

(*a*) Feed water is pumped into the boiler at the exhaust temperature t_2, against the full pressure p_1; in Fig. 55, the work of the feed pump could be represented by a very narrow rectangle against the line OA; in Fig. 56, the line EA represents the operation of imparting the heat $(q_1 - q_2)$, to raise the water from t_2 to t_1.

(*b*) Isothermal evaporation begins at A, and as the steam is pushed up from the surface of the water it pushes out the layer ahead of it, and so there is a continued transmission of pressure work until the moving piston is reached, and the external work of vaporization is finally done upon that moving surface. The steam is supposed to flow along the pipe without loss of pressure or of heat; that is, the pipe is assumed to be frictionless and nonconducting.

(*c*) The cylinder is taken to be perfectly nonconducting and nonabsorbent of heat, as in the Carnot engine; then after steam is cut

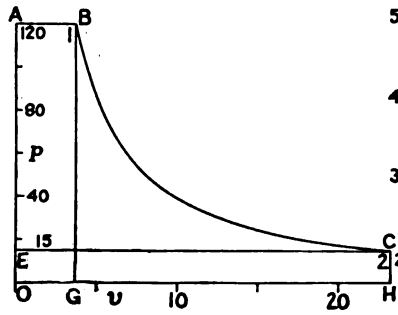

FIG. 55. — Pressure-volume Diagram for the Rankine Cycle.

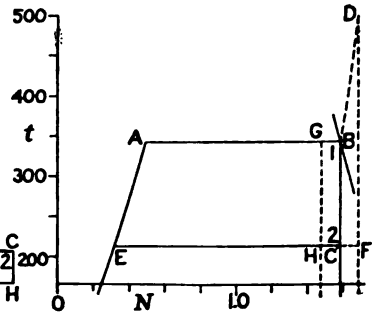

FIG. 56. — Temperature-entropy Diagram for the Rankine Cycle.

These diagrams are drawn with the same governing conditions as in Figs. 53 and 54; computations in Example 15.

off at B there is adiabatic expansion from B to C, carried clear down to the exhaust pressure.

(*d*) From C to E is represented the outflow of the steam from the cylinder and its condensation, or, in Fig. 56, the abstraction of heat by the condenser.

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The heat received, along EA and AB, is the total heat of formation above water at t_2; or, when the steam has entered the cylinder out to B, it carries the total heat h_1. Leaving the cylinder, from C, it carries out the total heat h_2 corresponding to the condition of the steam at that point. The input of energy is therefore

$$Q_1 = h_1 - q_2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (98)$$

while the output of work is $(h_1 - h_2)$; and the efficiency is given by the ratio

$$E = \frac{h_1 - h_2}{h_1 - q_2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (99)$$

Here h is used as a general symbol for total heat above water at 32 deg., leaving H for the tabular value of the total heat of dry steam; and in what follows k will be used in the same manner in relation to K.

Considering the work performance more in detail, we have in Fig. 55,

$$\text{work ABGO} = AP_1v_1;$$
$$\text{work BCHG} = I_1 - I_2 = k_1 - k_2, \text{ by Eq. (93)};$$
$$\text{work CEOH} = AP_2v_2.$$

Then the net work ABCE, in identity with the value above, is

$$AU = AP_1v_1 + k_1 - AP_2v_2 - k_2 \cdot$$
$$= h_1 - h_2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (100)$$

The use of the symbol h_1 instead of H_1 for the total heat at B implies that the cycle is not limited to the case where the steam is dry-saturated at cut-off. In either of the cycles which have been discussed, there may be partial evaporation at this point; and the Rankine cycle is more general than the Carnot, in that it covers also the use of superheated steam, since it does not require that the whole operation AB (Fig. 55) be isothermal, but merely that it be carried out under constant pressure.

Example 15. — With the pressure limits p_1 = 120 lb. and p_2 = 15 lb. find the input, output, and efficiency of the Rankine cycle for three cases, namely,

(*a*) With dry steam or complete evaporation at full admission, point B, Fig. 55 or Fig. 56;

(*b*) For steam with 10 per cent of moisture;

(*c*) For steam superheated to 500 deg. fahr.

Cases (*b*) and (*c*) are represented by the dotted lines at the right-hand end of the diagram in Fig. 56.

All the tabular values needed for this problem are collected in Example 14, and for case (*a*) the important quantities here required have been worked out; the changes in point designation are indicated in the tabulation below by bracketing the symbols used in Example 14.

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For case (*b*), with entropy b_1 equal to 1.0957, the presence of 10 per cent of moisture will diminish the total entropy, out to the adiabatic line, by 0.1096; and with $b_2 = 1.4407$ this will increase m_2 by the amount $0.1096 \div 1.4407 = 0.0761$, making it $0.1156 + 0.0761 = 0.1917$. The change in total heat h_1 will be $0.1 \times 877.6 = 87.8$, and that in h_2 will be $0.0761 \times 969.0 = 73.7$: the table contains the resulting quantities.

For case (*c*), from Table VII total heat $h_1 = 1275.7$, and from Table VIII entropy $n_1 = 1.686$; this exceeds the saturation value 1.588 by 0.098. At the lower pressure, 0.098 in 1.441 decreases m_2 by 0.068, making it 0.048, and increases h_2 by $0.068 \times 969.0 = 65.9$; these changes taking place from the values in case (*a*).

The results are tabulated in condensed form. As compared with Example 14, case (*a*) shows a drop in efficiency from 0.1602 to 0.1538. With the rectangular temperature-entropy diagram in Fig. 54, cutting off a piece at the right end (because of incomplete vaporization) would not change the efficiency; here it is lowered by the amount shown in case (*b*). In case (*c*) there is some rise in efficiency as compared with case (*a*), but because the increase in tempera-

Case (*a*) $m_1 = 0$	Case (*b*) $m_1 = 0.10$	Case (*c*) $t_1 = 500°$
$h_1 = 1189.8$ (h_2)	1102.0	1275.7
$h_1 - q_2 = 1008.8$	921.0	1094.7
$m_2 = 0.1156$ (m_2)	0.1917	0.048
$h_2 = 1038.1$ (h_2)	964.4	1104.0
$h_1 - h_2 = 151.7$	137.6	171.7
$E = 0.1538$	0.1494	0.1568

ture range (through extension above the line AB, Fig. 56) affects only a small portion of the heat handled, the change is not very great.

(*e*) THE STEAM ENGINE CYCLE. — Having passed from the Carnot to the Rankine cycle because of a fundamental change in the arrangement of apparatus, the next step to be made will cover the change from an ideal, thermally neutral cylinder to the actual, heat-absorbing and conducting metal cylinder.

When the steam enters the cylinder it comes into contact with containing surfaces which have been cooled by exposure to the exhaust during the cycle just completed, and which have a high capacity for heat. During the admission period, then, a part of the steam is condensed, its latent heat going into the metal of the cylinder and piston; it therefore shrinks in specific volume, one pound filling the space AB instead of AM in Fig. 57 (AM here is the same as AB in Fig. 55). With a given cylinder, this means that a greater weight of steam will be taken in during admission than would be needed if the steam remained in the state in which it comes from the boiler.

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The absorption of heat by the metal surfaces may continue into the first part of expansion; but as the pressure drops the temperature gradient soon reverses, and heat flows back into the steam, at an increasing rate. The result is, that instead of condensation to supply heat for external work, there is usually some reëvaporation of moisture. Of the heat thus returned to the steam, a small (and as the temperature of return falls, a decreasing) portion is converted into work; but that which flows into, and is carried off by, the exhaust steam is altogether wasted. For this latter quantity, the action is equivalent to what was described in § 10 (*h*), inasmuch as it is a mere transfer of heat from source to receiver, without performance of any useful work.

Because of the supply of heat which the steam receives from the cylinder as the pressure is lowered, the expansion curve usually approximates quite fairly to the curve $pv = C$ — this fact being wholly a matter of experience, not deducible by *a priori* reasoning. Of course, the exact form of the curve varies considerably with changing conditions, the study of which belongs to the next chapter. To get what we shall call the ideal steam diagram, we now assume that the equilateral hyperbola may be used as curve of expansion.

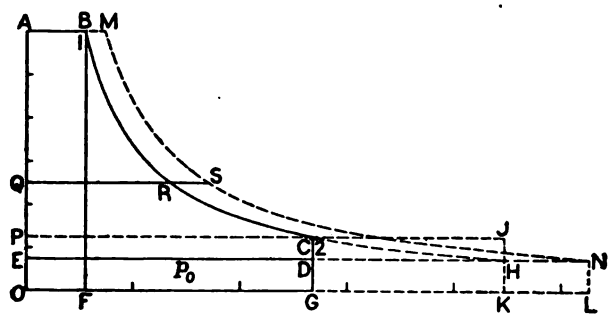

FIG. 57. — The Steam-engine Cycle, in Comparison with the Rankine Cycle. See § 15 (*e*) and Example 16.

(*f*) THE IDEAL STEAM DIAGRAM. — In Fig. 57, the effective initial volume AB is determined, from the full volume AM, by the percentage or fraction of initial condensation, for which ratio AM is the base. The curve BCH is of the form $pv = C$, according to the discussion in the last article. Expansion is not carried to the exhaust pressure at H, but is stopped at some earlier point C, and the line CD shows the operation of release. This is taken as equivalent to cooling at constant volume when laying out the temperature-entropy diagram, Fig. 58. The exhaust, from D to E, is complete, or all the steam is expelled from the cylinder; in other words, the engine has no clearance space. In the present chapter we shall not go beyond this simple diagram. Its further

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modification by secondary influences, such as pipe and valve losses (of pressure), clearance and compression, etc., will be taken up in the next chapter, with the study of actual indicator diagrams.

(*g*) INCOMPLETE EXPANSION. — There are two good reasons why it is advantageous to stop the expansion at C, Fig. 57, instead of carrying it to H. The size of an engine cylinder is determined by the necessity of its holding all the steam at the release volume OG. In comparing steam with air as a medium for use in the Carnot engine — see Art. (*b*) and § 8 (*h*) — the advantage of a high mean effective pressure and a small piston displacement has been well shown and strongly emphasized. With this idea in mind, it appears highly undesirable to make the cylinder volume, referred to the pound of steam, equal to OK instead of OG, merely to save the small work area CHD.

Just what will be the best terminal pressure, at C, is a question to be settled by a judicious balancing of the evils of loss of available work on one hand and a needlessly big and costly engine on the other. The second reason for using incomplete expansion, now to be set forth, has a decided and definite influence in this connection.

Suppose that all the frictional resistances in the machine are bunched together and reduced to an equivalent mean pressure on the piston, measured in the same terms as the steam pressure p — see § 28 (*g*) for some results from such a reduction. Let this be added to the back pressure OE, raising it say to OP. Then if the expansion be carried from G to K, the positive work of steam on piston will be CHKG, the negative work of piston on steam and against friction will be JCGK, and there will be a net loss equal to area JCH. It appears then that there is an actual loss of output in carrying expansion beyond the point where terminal pressure equals back pressure plus engine friction. This is the limit of economical expansion, and the considerations described in the first part of this article tend to keep actual working well within this limit. The terminal pressure varies with kind of engine and condition of service. In noncondensing engines it ranges from 25 to 40 lb. absolute, in condensing engines it is usually from 8 to 10 lb. These are normal values, for rated load; the pressure falls with underload, rises with overload.

(*h*) THE TEMPERATURE-ENTROPY DIAGRAM for the steam-engine cycle is given in Fig. 58. Broken line EAM represents the combined operations of water heating and evaporation, in the boiler; MB shows the initial absorption of heat by the cylinder walls. Expansion line BC corresponds to the equilateral hyperbola in Fig. 57, and its slant toward the right is due to heat taken up from the metal surfaces, which in amount is equal to the area under BC, between vertical ordinates from

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B and C. In general, the method of locating a point like R on the curve BC of Fig. 58 is to make the ratio QR/QS the same here as on Fig. 57. This can be done graphically, but measurement and slide rule computation is generally less trouble than pure geometrical construction. Line CD represents cooling at constant volume (see Fig. 48), line DE plain condensation at constant pressure and temperature. The wasted work, or the portion of the ideally available energy which the engine fails to convert into work, is the area MBCDN. If we imagine the heat receiver to be at the exhaust temperature t_2, and assume that the waste

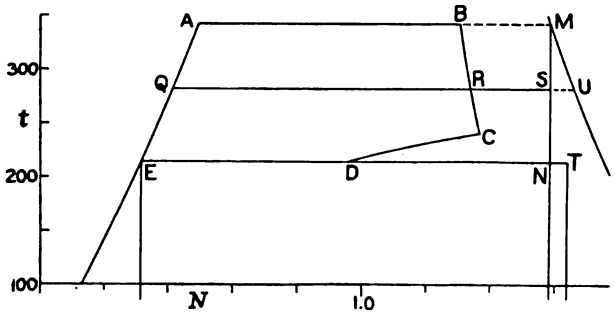

FIG. 58. — Thermal Diagram of the Steam-engine Cycle. Proportions correspond to Fig. 57.

heat finally settles into the receiver at that temperature, it will increase the entropy acquired by the receiver from EN to ET — compare § 10 (*c*).

(*i*) WORK PER POUND OF STEAM. — If we know the probable amount of initial condensation within the cylinder, and can thus fix the effective volume AB in Fig. 57, it is easy to calculate the work per pound of steam in the steam-engine cycle, for given pressure limits. The data are, boiler pressure p_1, exhaust pressure p_0, and either the terminal pressure p_2 or the ratio of expansion $r = v_2/v_1$. It is desirable to express the effective work in terms of p_1, p_0, and v_2.

From Fig. 57 the component work quantities are:

$$\text{area ABFO} = p_1 v_1 = p_1 \times v_2 \times \frac{1}{r};$$

$$\text{area BCGF} = p_1 v_1 \log_e r = p_1 v_2 \frac{\log_e r}{r};$$

$$\text{area DEOG} = p_0 v_2.$$

Then the net work in foot pounds, with pressure values in pounds per square inch, is

$$U = 144\, v_2 \left[p_1 \frac{1 + \log_e r}{r} - p_0 \right] \quad . \quad . \quad . \quad . \quad . \quad (101)$$

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The expression in brackets is the mean effective pressure of the diagram. Analyzed, it is the difference between the mean total pressure during the forward stroke,

$$p_{mf} = p_1 \frac{1 + \log_e r}{r}, \quad \ldots \ldots \quad (102)$$

and the mean back pressure p_{mb} or p_0 in the return stroke.

In designing an engine for a given power requirement, the first thing to do is to fix the size of the cylinder, and the first determinant called for is the mean effective pressure that can be expected. The simplest procedure is to compute a value with the help of Eq. (102), and modify it by an empirical "diagram factor" which takes account of the losses due to the secondary actions that have not yet been considered. Further development of this matter will be found in § 29 (*b*).

EXAMPLE 16. — In Fig. 57, $p_1 = 120$ lb., $p_2 = 25$ lb., $p_0 = 15$ lb., $x_1 = 0.75$. Find work per pound of steam, mean effective pressure, and thermal efficiency.

In Example 14, $s_1 = 3.727$; then v_1 here is 2.800 cu. ft., by Eq. (58).

The ratio r is equally well got from pressures, or it is $120 \div 25 = 4.8$; therefore $\log_e r = 2.3026 \log r = 2.3026 \times 0.68124 = 1.5686$; and

$$\frac{1 + \log_e r}{r} = \frac{2.5686}{4.8} = 0.53513.$$

Now $p_{mf} = 120 \times 0.5351 = 64.21$ lb.,

and $p_m = 64.21 - 15 = 49.21$ lb. per sq. in.

Volume $v_2 = 2.800 \times 4.8 = 13.440$ cu. ft.

The effective work is

$$U = 144 \times 49.21 \times 13.44 = 95{,}240 \text{ ft. lb.}$$

In heat units this is,

$$AU = 95{,}240 \div 778 = 122.4 \text{ B.t.u.}$$

The engine is supposed to have received dry steam, so that the problem is a continuation of case (*a*) in Example 15. There the heat supplied was found to be $Q_1 = 1008.8$ B.t.u., and the Rankine cycle efficiency was $E_I = 0.1538$ — compare Eqs. (41) to (43), § 8 (*g*), for the different kinds of efficiency.

Here then the absolute efficiency is

$$E_A = 122.4 \div 1008.8 = 0.1213.$$

The relative efficiency, which corresponds to area ABCDE ÷ area AMNE in Fig. 58, is therefore

$$E_R = \frac{0.1213}{0.1538} = 0.789.$$

For well-designed engines in good condition, the actual value of E_R ranges from 0.6 to 0.7 — see Table 13, page 268.

(*j*) THE REGENERATIVE CYCLE. — A scheme for approximating the Carnot cycle in effect, used in some high-grade steam plants which

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have to meet exceptional requirements as to thermal economy, is illustrated in elementary fashion in Fig. 59. Going back to § 8 (c), and retaining the conditions of receiving all heat from without at T_1 and rejecting all heat to without at T_2, it must be remarked that the adiabatic is not the only possible process for the operations in which the temperature of the medium is raised and lowered. If in the temperature-lowering process some heat is abstracted and, without drop in temperature, carried over into the temperature-raising process, less heat will be rejected and less will have to be supplied.

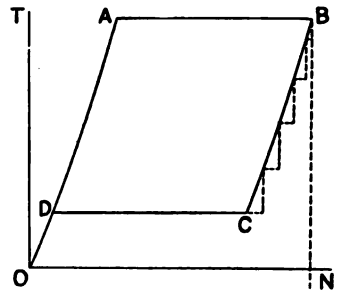

FIG. 59. — The Regenerative Cycle, Ideal Diagram.

In its perfection, the scheme requires that the "expansion" (in the ordinary steam-engine sense) follow on the temperature-entropy diagram the line BC, Fig. 59, exactly like DA; and the heat abstracted in this subadiabatic expansion is used to raise the feed water to the upper temperature t_1. Actually, the heat abstraction takes place at several distinct points along the line of temperature drop, giving to the ideal operation (in a nonconducting engine) the stepped outline dotted on Fig. 59. This diagram corresponds to a quadruple-expansion engine, in which steam was withdrawn from each of the three receivers and from the low-pressure cylinder at release — see Fig. 140, § 27 (i). A series of feed-water heaters is provided, with pumps between them, and the water is raised in pressure and temperature coincidently.

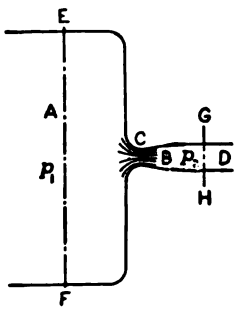

FIG. 60. — Conditions of Jet Formation.

§ 16. The Dynamic Force Cycle

(a) THE STEAM JET. — The conditions for the formation of a perfect steam jet are illustrated in Fig. 60. The large vessel A is filled with steam at the pressure p_1, which flows toward the outlet at a very low velocity V_1. In entering and flowing through the nozzle B, the steam drops rapidly in pressure and gains in velocity; the work that would be done against the piston in an engine is here used in accelerating the steam itself, or is changed into kinetic energy of the jet. In the tube D the fully established jet has a high velocity V_2 and throughout the steam substance there exists the stress or pressure p_2. We assume that the material of the confining surfaces is thermally neutral, and that

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these surfaces are frictionless. Under these assumptions, the expansion within the nozzle must be adiabatic and isentropic, for no heat can be received or lost, and there will be none of the throttling effect described in § 17 (*d*).

(*b*) ENERGY OF THE JET. — If the pressure or stress within the steam is plotted on instantaneous specific volume as base, the resulting diagram, Fig. 61, will be exactly like the Rankine cycle, Fig. 55. The amount of work done upon the steam mass and stored in kinetic form is determined by the following line of reasoning:

As a pound of steam is pushed past any dividing plane like EF, Fig. 60, it receives the static-pressure work p_1v_1 from the steam behind it, and as it pushes past a dividing plane like GH it performs the work p_2v_2 upon the steam ahead of it. In the drop from p_1 to p_2 it loses the internal energy $(I_1 - I_2)$, which is equivalent to the area BCVG in Fig. 61. External work p_1v_1 plus internal energy I_1 is the total heat h_1 coming into the nozzle; external work p_2v_2 plus internal energy I_2 is the total heat h_2 going out of the nozzle; and since no energy is interchanged with any outside body, the difference $(h_1 - h_2)$ must be equal to the kinetic energy gained. Disregarding the approach velocity V_1, which is of insignificant value, and using plain V instead of V_2, the equation for the kinetic jet energy per pound of steam is

$$E = \frac{V^2}{2g} = 778\,(h_1 - h_2) \text{ ft. lb.} \quad . \quad . \quad . \quad . \quad . \quad (103)$$

It is of interest to note that, whereas $p\,dv$ is the element of static-pressure work (see Fig. 31), the corresponding element of jet energy is $v\,dp$, as indicated on Fig. 61. Briefly, suppose a slice of the nozzle space to be set off by means of a pair of cross planes exceedingly close together. At one plane the pressure is p, at the other $(p - dp)$. The pound of steam having the volume v at pressure p, the unbalanced pressure dp acts through the volume or displacement v, and does the work $v\,dp$ in producing the acceleration dV/dt — this t being time, not temperature.

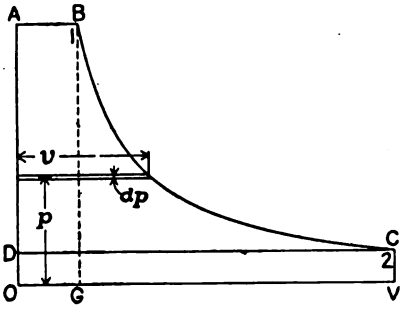

FIG. 61. — Pressure-volume Diagram for the Jet.

(*c*) CROSS AREA OF THE JET. — The method of calculating the energy E, Eq. (103), has been fully exemplified in § 15 (*d*). To get the velocity V at any condition p_2, we need only make the calculation,

$$V = \sqrt{64.32 \times 778\,(h_1 - h_2)} = \sqrt{50{,}040\,(h_1 - h_2)} = 223.7\,\sqrt{h_1 - h_2}. \quad (104)$$

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We shall take the unit jet to be that which discharges one pound of steam in one second of time. If we get the specific volume at p_2, which will be $v_2 = x_2 s_2$, and note that this is in cubic feet while V is in feet per second, we find the cross area of the unit jet in square inches to be

$$a = 144 \frac{v_2}{V}. \quad \ldots \ldots \ldots \quad (105)$$

EXAMPLE 17. — If steam initially dry at 120 lb. absolute expands in a perfect unit jet to 15 lb. absolute, what will be its energy, velocity, and area of cross section?

In Example 15, case (*a*), for these data, h_1 and h_2 have been computed as 1189.8 and 1038.1 B.t.u., respectively, and $(h_1 - h_2)$ as 151.7 B.t.u. The kinetic energy is therefore

$$E = 151.7 \times 778 = 118{,}020 \text{ ft. lb.}$$

The velocity is

$$V = \sqrt{64.32 \times 118{,}020} = 2755.2 \text{ ft. per second.}$$

With the final moisture-fraction $m_2 = 0.1156$, the volume v_2 is $26.27 - 3.04 = 23.23$ cu. ft., as in Example 14; then the area of the jet at 15 lb. pressure is found to be

$$a = \frac{144 \times 23.23}{2755} = \frac{3345}{2755} = 1.214 \text{ sq. in.}$$

(*d*) FORM OF THE JET. — If, after the manner of Example 17 and with a given initial pressure p_1, the area a be computed for a series of values of the instantaneous lower pressure p, a curve of area relative to pressure can be laid out, as in Fig. 62. Following the curve ABCDE, we see first a rapid decrease in area, shown by AB and due to the rapid growth of velocity. Presently, as the steam expands, specific volume gains on velocity, and the two keep well together through a long range of pressure, as is shown by the small variation in a from B to D. At the low-pressure end of the operation, volume grows much faster than velocity, hence the sharp rise in a along DE. At C the area has a minimum value, and this may well be called the throat of the jet; it is located at just 0.58 of the initial pressure. The fact that a jet of steam enlarges after it passes the throat pressure is the reason why an enlarging or flaring nozzle must be used in order to get the full effect of a wide range of pressure drop.

The limiting ordinates in Fig. 62, at the left for $p = p_1$, at the right for $p = 0$, are asymptotes to the a curve, calling for infinite values of a. The first condition implies zero velocity, which can never exist in a plant at work, because steam must flow, as from boiler to turbine or engine; still, steam-pipe velocities are so low compared with those in turbine jets that they correspond to points far up on the A end of the curve. At zero pressure, or for the ideal case of a jet flowing into a

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perfect vacuum, while velocity would be finite, volume would be infinite, hence area would have to be infinite. In steam-turbine practice, a good working "vacuum" is in the neighborhood of 1 lb. per sq. in. absolute, and sometimes it even drops below 0.5 lb.

In general form, the actual jet agrees with the ideal, but quantitatively there is some discrepancy, due chiefly to failure to realize completely the assumption that all the available heat energy is converted into kinetic energy of the jet. For the study and determination of the secondary, wasteful actions, it is necessary to compare the observed with the ideal jet; and after the degree of modification is known, it can be used to adjust values computed for ideal conditions. With these uses in view, it is important that the proportions of the ideal jet be in convenient shape for ready reference and comparison.

(*e*) THE STEAM-JET TABLES. — The proportions of the isentropic jet from initial dry steam are fully given in Table 6. As noted in the heading, this table represents exactly the jet which starts at 120 lb. absolute pressure. By means of the factors in Table 7, the numbers in Table 6 can be modified so as to fit jets from other initial pressures. The whole scheme is based on the assumption — a good approximation, although by no means rigorously exact — that in terms of pressure-ratio the jets are similar, in the geometrical sense of the word. The tabular quantities will now be defined and their use illustrated, and then the question of the real degree of similarity among jets from different pressures will be more fully gone into.

Pressure ratio p/p_1: This ratio of variant falling pressure to fixed initial pressure is the fundamental determinant, used as "argument" in Table 6 and as base in Figs. 62 to 65. The numbers in the other columns of Table 6 give the result of expansion from p_1 to p.

Energy E and energy ratio R_E: In col. 2 of Table 6, the jet energy in B.t.u. is the difference $(h_1 - h_2)$, as explained in Art. (*b*). For any other initial pressure than 120 lb., multiply this E_{120} by R_E from Table 7, and the result will be E for the same ratio of expansion from the other pressure p_1.

Velocity V and velocity ratio R_V: In col. 3 of Table 6, the jet velocity in feet per second is derived from E by Eq. (104). For any other p_1, multiply this V_{120} by R_V from Table 7; note that R_V is simply the square root of R_E.

Volume ratio v/v_1: In col. 4 of Table 6, this gives the exact form of the adiabatic curve from dry steam at 120 lb. initial pressure. How closely this column of ratios will fit curves from other pressures can be judged by reference to Fig. 50.

Area ratio a/a_0: This shows the form of the jet, after the manner of

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the curves in Fig. 62, but the measurement is made relative instead of absolute by giving any area a in terms of the throat area a_0.

Area ratio R_a and throat area a_0: R_a is the ratio of the sizes of different jets, showing how the jet from p_1 compares in cross area with that from 120 lb. It is got by dividing any a_0 in col. 5 of Table 7 by the particular value 0.5868 for 120 lb. The two columns are strictly in parallel, a_0 being the absolute value at the throat, in square inches for the pound-per-second jet, while R_a shows relative values, not only at the throat but at any stage in the pressure drop.

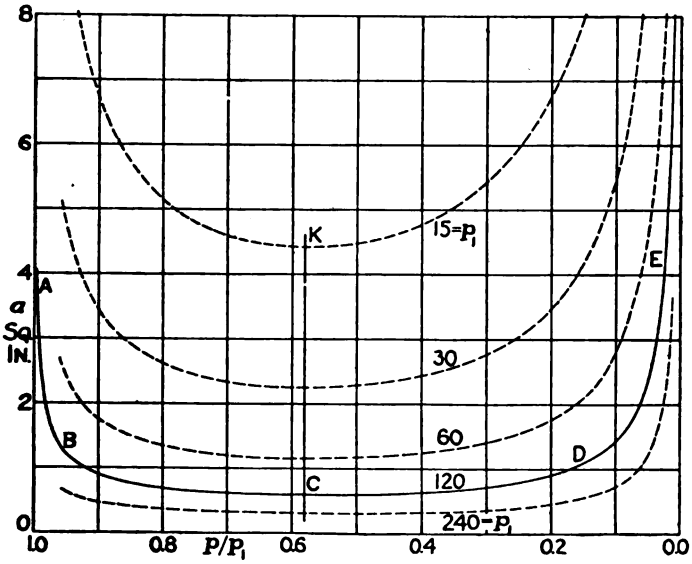

FIG. 62. — Curves of Jet Area.

Base, ratio of variant lower pressure to initial pressure; ordinate, cross area of ideal unit jet. Jets start from initial dry steam at the five (absolute) pressures marked on the curves.

Napier divisor D and its ratio of variation, R_D: These are explained in Art. (*h*), where the help which D gives in smoothly interpolating a_0 and R_a is also described.

EXAMPLE 18. — For an ideal jet from $p_1 = 200$ lb. to $p_2 = 60$ lb., find by means of the tables the final energy, velocity, and cross area.

The pressure ratio p/p_1 is 0.3; for this datum we get from Table 6,

$$E = 92.33, \qquad V = 2150, \qquad a = 1.224 a_0.$$

From Table 7, with $p_1 = 200$ lb., the energy ratio R_E is 1.0240; this means that the energy of the 200-lb. jet is 2.4 per cent greater than that of the 120-lb. jet; the value is therefore,

$$E = 92.33 \times 1.0240 = 92.33 + 2.22 = 94.55 \text{ B.t.u.}$$

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TABLE 6. PROPORTIONS OF THE IDEAL STEAM JET

Computed for initial dry saturated steam at 120 lb. per sq. in. absolute pressure.

1	2	3	4	5	1	2	3	4	5
Ratio of pressures.	Energy of jet, B.t.u.	Velocity of jet, feet per sec.	Ratio of volumes.	Ratio of areas.	Ratio of pressures.	Energy of jet, B.t.u.	Velocity of jet, feet per sec.	Ratio of volumes.	Ratio of areas.
$\frac{p}{p_1}$	E	V	$\frac{v}{v_1}$	$\frac{a}{a_0}$	$\frac{p}{p_1}$	E	V	$\frac{v}{v_1}$	$\frac{a}{a_0}$
0.00	0.00	0	1.000	.....	0.60	40.80	1429	1.564	1.0015
0.99	0.84	205	1.009	4.512	0.59	42.09	1451	1.587	1.0005
0.98	1.67	289	1.018	3.221	0.58	43.40	1474	1.611	1.000
0.97	2.51	354	1.027	2.651	0.57	44.74	1496	1.636	1.0002
0.96	3.36	410	1.037	2.311	0.56	46.11	1519	1.662	1.0007
0.95	4.22	460	1.046	2.082	0.55	47.50	1541	1.689	1.002
0.94	5.09	505	1.056	1.913	0.54	48.91	1564	1.716	1.003
0.93	5.96	546	1.066	1.784	0.53	50.34	1587	1.744	1.005
0.92	6.84	585	1.076	1.682	0.52	51.80	1610	1.774	1.008
0.91	7.73	622	1.086	1.598	0.51	53.29	1633	1.804	1.011
0.90	8.63	657	1.097	1.527	0.50	54.80	1656	1.836	1.014
0.89	9.54	691	1.107	1.466	0.49	56.33	1679	1.869	1.018
0.88	10.46	724	1.118	1.414	0.48	57.88	1702	1.903	1.023
0.87	11.39	755	1.129	1.368	0.47	59.46	1725	1.938	1.028
0.86	12.33	785	1.141	1.328	0.46	61.07	1748	1.975	1.034
0.85	13.28	815	1.153	1.293	0.45	62.72	1772	2.014	1.040
0.84	14.24	844	1.165	1.262	0.44	64.40	1795	2.054	1.047
0.83	15.21	872	1.177	1.235	0.43	66.11	1819	2.096	1.054
0.82	16.18	900	1.190	1.210	0.42	67.86	1843	2.140	1.062
0.81	17.17	927	1.203	1.187	0.41	69.65	1867	2.186	1.071
0.80	18.17	953	1.216	1.166	0.40	71.48	1891	2.234	1.080
0.79	19.18	979	1.229	1.147	0.39	73.34	1916	2.284	1.090
0.78	20.19	1005	1.243	1.131	0.38	75.24	1940	2.337	1.101
0.77	21.21	1030	1.257	1.116	0.37	77.19	1965	2.392	1.113
0.76	22.24	1055	1.271	1.103	0.36	79.19	1991	2.450	1.126
0.75	23.29	1080	1.286	1.091	0.35	81.24	2016	2.511	1.139
0.74	24.35	1104	1.301	1.079	0.34	83.34	2042	2.576	1.154
0.73	25.43	1128	1.317	1.068	0.33	85.50	2068	2.644	1.170
0.72	26.53	1152	1.333	1.058	0.32	87.72	2095	2.717	1.187
0.71	27.64	1176	1.350	1.050	0.31	89.99	2122	2.795	1.205
0.70	28.77	1200	1.367	1.042	0.30	92.33	2150	2.877	1.224
0.69	29.91	1224	1.384	1.035	0.29	94.74	2178	2.964	1.245
0.68	31.06	1247	1.402	1.029	0.28	97.22	2206	3.057	1.268
0.67	32.22	1270	1.420	1.023	0.27	99.79	2235	3.156	1.292
0.66	33.40	1293	1.439	1.018	0.26	102.45	2264	3.263	1.319
0.65	34.59	1316	1.458	1.014	0.25	105.21	2294	3.378	1.347
0.64	35.80	1338	1.478	1.010	0.24	108.07	2325	3.502	1.378
0.63	37.03	1361	1.499	1.007	0.23	111.03	2357	3.636	1.411
0.62	38.27	1384	1.520	1.005	0.22	114.08	2390	3.781	1.447
0.61	39.53	1406	1.542	1.003	0.21	117.27	2423	3.939	1.487

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TABLE 6. PROPORTIONS OF THE IDEAL STEAM JET—*Continued.*

1	2	3	4	5	1	2	3	4	5
Ratio of pressures.	Energy of jet, B.t.u.	Velocity of jet, feet per sec.	Ratio of volumes.	Ratio of areas.	Ratio of pressures.	Energy of jet, B.t.u.	Velocity of jet, feet per sec.	Ratio of volumes.	Ratio of areas.
$\frac{p}{p_1}$	E	V	$\frac{v}{v_1}$	$\frac{a}{a_0}$	$\frac{p}{p_1}$	E	V	$\frac{v}{v_1}$	$\frac{a}{a_0}$
0.20	120.59	2457	4.113	1.532	0.04	220.70	3323	17.21	4.737
0.19	124.06	2492	4.305	1.580	0.035	228.23	3380	19.40	5.250
0.18	127.70	2528	4.516	1.634	0.03	236.81	3443	22.33	5.934
0.17	131.53	2566	4.749	1.694	0.025	246.78	3514	26.23	6.827
0.16	135.57	2605	5.010	1.760					
					0.02	258.66	3598	32.06	8.15
0.15	139.83	2645	5.305	1.835	0.018	264.26	3637	35.28	8.87
0.14	144.35	2688	5.639	1.919	0.016	270.40	3679	39.17	9.74
0.13	149.17	2732	6.023	2.016	0.014	277.29	3725	44.21	10.86
0.12	154.33	2779	6.469	2.128					
0.11	159.84	2829	6.988	2.260	0.012	285.08	3777	50.81	12.53
					0.010	294.19	3837	59.69	14.23
0.10	165.89	2881	7.60	2.413	0.008	305.13	3908	75.40	17.65
0.09	172.48	2938	8.35	2.599	0.006	318.60	3993	95.16	21.80
0.08	179.76	2999	9.27	2.827					
0.07	187.87	3066	10.43	3.114	0.004	337.48	4110	137.9	30.69
0.06	197.11	3141	11.97	3.487	0.003	350.55	4189	178.9	39.07
					0.002	368.49	4294	258.9	55.15
0.05	207.84	3225	14.09	3.999	0.001	396.71	4456	488.6	100.29
0.045	213.85	3271	15.49	4.331					

The velocity ratio being $R_V = 1.0119$, the new velocity is

$$V = 2150 \times 1.0119 = 2150 + 26 = 2176.$$

The area a can be got in several ways. Using $a/a_0 = 1.224$ from Table 6 with $a_0 = 0.3569$ for 200 lb. from Table 7, we have

$$a = 1.224 \times 0.3569 = 0.4368 \text{ sq. in.}$$

Or, with $a_0 = 0.5868$ and $a/a_0 = 1.224$, the area of the 120-lb. jet at $p/p_1 = 0.3$ is

$$a_{120} = 1.224 \times 0.5868 = 0.7182 \text{ sq. in.;}$$

and with the ratio $R_a = 0.6081$ for 200 lb. we have

$$a = 0.6081 \times 0.7182 = 0.4368 \text{ sq. in.}$$

Evidently, the first method is the simpler and more direct.

The manner in which the computation of E and V has just been indicated shows the proper way to use the ratios from Table 7 with

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TABLE 7. FACTORS FOR DIFFERENT INITIAL PRESSURES

1	2	3	4	5	6	7
Initial pressure.	Ratio of energies.	Ratio of velocities.	Ratio of cross areas.	Area at throat, square inches.	Napier divisor.	Ratio of variation of D.
p_1	R_E	R_V	R_a	a_0	D	R_D
250	1.0326	1.0162	0.4892	0.2871	71.77	1.0192
240	1.0311	1.0154	0.5091	0.2988	71.70	1.0182
230	1.0295	1.0146	0.5307	0.3114	71.62	1.0171
220	1.0278	1.0138	0.5542	0.3252	71.54	1.0159
210	1.0260	1.0129	0.5799	0.3403	71.46	1.0148
200	1.0240	1.0119	0.6081	0.3569	71.37	1.0135
190	1.0217	1.0108	0.6392	0.3752	71.27	1.0121
180	1.0192	1.0096	0.6738	0.3954	71.17	1.0107
170	1.0166	1.0083	0.7123	0.4180	71.06	1.0091
160	1.0138	1.0069	0.7556	0.4434	70.94	1.0074
150	1.0108	1.0054	0.8046	0.4721	70.82	1.0057
140	1.0075	1.0037	0.8605	0.5049	70.69	1.0039
130	1.0039	1.0019	0.9250	0.5428	70.56	1.0020
120	1.0000	1.0000	1.0000	0.5868	70.42	1.0000
110	0.9956	0.9978	1.0882	0.6386	70.25	0.9976
100	0.9904	0.9952	1.194	0.7006	70.06	0.9949
95	0.9878	0.9939	1.255	0.7364	69.96	0.9935
90	0.9850	0.9925	1.323	0.7761	69.85	0.9919
85	0.9819	0.9909	1.398	0.8205	69.74	0.9903
80	0.9786	0.9892	1.483	0.8703	69.62	0.9886
75	0.9750	0.9874	1.579	0.9265	69.49	0.9868
70	0.9712	0.9855	1.688	0.9907	69.35	0.9848
65	0.9671	0.9834	1.814	1.0645	69.19	0.9825
60	0.9627	0.9812	1.960	1.1503	69.02	0.9801
55	0.9579	0.9787	2.134	1.2525	68.84	0.9775
50	0.9526	0.9760	2.340	1.3728	68.64	0.9747
45	0.9468	0.9730	2.591	1.5204	68.42	0.9716
40	0.9401	0.9696	2.904	1.7040	68.16	0.9680
35	0.9327	0.9658	3.304	1.9390	67.87	0.9638
30	0.9242	0.9613	3.836	2.2513	67.54	0.9591
25	0.9138	0.9559	4.579	2.6868	67.17	0.9538
20	0.9008	0.9491	5.687	3.3375	66.75	0.9478
15	0.8840	0.9402	7.523	4.4147	66.22	0.9403
10	0.8610	0.9279	11.155	6.546	65.46	0.9295
5	0.8275	0.9097	21.86	12.827	64.13	0.9107

the slide rule: that is, do not multiply the numbers from Table 6 by the whole ratio, but only by the difference between the ratio and unity; then add (or subtract) this result to get the adjusted value sought.

(*f*) SIMILARITY OF STEAM JETS. — The five curves drawn on Fig. 62

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are intended to cover the whole range of common steam turbine practice. We shall now make a close comparison among these five jets, from initial dry steam at 240, 120, 60, 30, and 15 lb. absolute, with the especial purpose of testing the accuracy of the method of Tables 6 and 7. The quantities to be considered are, energy (which covers velocity), specific volume, and area; and of these, specific volume has been fully covered by § 14 (*d*), Table 5, and Fig. 50.

In Fig. 63 the energy ratio R_E is determined and interpolated. First of all, for each of the jets to be compared with that from 120 lb., there

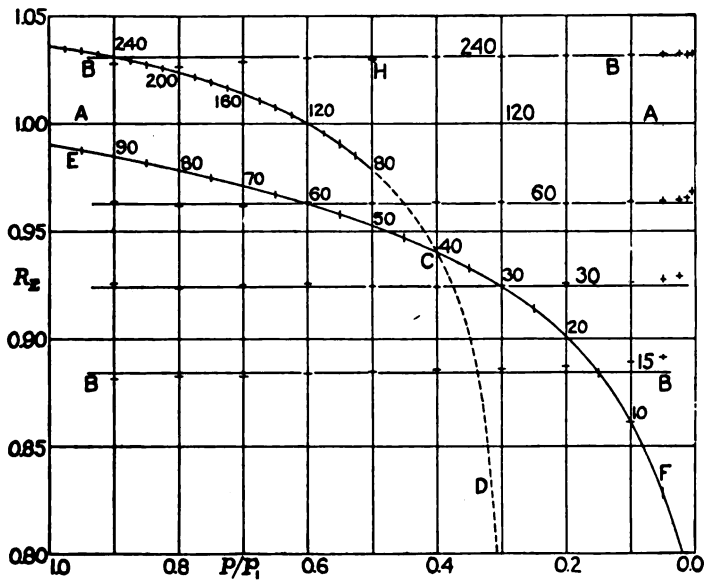

FIG. 63. — Comparison of Jet Energies, and Determination of Energy Ratio R_E.

For lines AA and BB, same base as in Fig. 62; for curves BCD, ECF, base is initial pressure p_1 (col. 1 of Table 7) to scales marked along curves. Ordinate is R_E.

was computed a series of values of this ratio. Thus at $p/p_1 = 0.5$, the energy of the 120-lb. jet is 54.80 B.t.u. (see Table 6), and that of the 240-lb. jet is 56.44 B.t.u., both being calculated as indicated in Example 17, but without reduction to foot pounds. Then $R_E = 56.44 \div 54.80 = 1.0299$, and this value is plotted at the point H on Fig. 63. The short cross marks on the ordinate lines, keeping close to the BB lines, show the results of the calculations. These horizontal lines of mean ratio are drawn at the average height of the plotted points (except the rapidly rising points at the right-hand end of the lower curves). The departure of the points from the mean lines measures the error in the

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assumption of a constant R_E between the jet from p_1 lb. and that from 120 lb., and in the use of Table 7. The small discrepancies between the plotted points and fair curves represent irregularities in computation of the order of 0.01 to 0.03 B.t.u., and are due to the accumulation of fractional errors in the last figure of the steam table numbers. Disregarding these, we may say that the error in using a constant R_E for the whole range of any jet does not exceed one-fourth of one per cent., except at the low-pressure end of the lower jets, where it rises to 0.5 per cent by the time a pressure of 1 lb. absolute is reached.

The curves BCD and ECF, really the same curve to two scales, represent R_E as given in col. 2 of Table 7. After the BB lines had been fixed, the curve was drawn through the five points thus located, and the intermediate tabular values were measured from it.

(*g*) COMPARISON OF JET AREAS. — In Fig. 64, the group of curves near the line MN shows by how much the other four jets depart from similarity with that which begins at 120 lb. pressure. The comparison lies between computed and tabular values of the ratio a/a_0; that is, between values computed for the other jets and those tabulated for the standard, 120-lb. jet. As an example, at $p = 0.2\,p_1$, the area of the 30-lb. jet was computed as 3.467 sq. in., while a_0 for this jet is 2.251; then the ratio of jet proportion is $a/a_0 = 1.540$. For the 120-lb. jet, the corresponding ratio is 1.532, by Table 6. If the jets were truly similar, these proportion factors would be equal; to show the degree of actual inequality, the ratio between them, $1.540 \div 1.532 = 1.0052$, is plotted in Fig. 64. From another point of view, it is as if the jet curves in Fig. 62 were brought together by means of factors — the factor a constant for any jet, and the reciprocal of R_a in Table 7 — which would give them exactly the same ordinate at the throat, and ratios were then taken between the slightly differing ordinates at other points on the scale of pressure drop.

In respect to form, as thus expressed in the manner of variation of cross area, the disagreement among the jets is far greater than in Fig. 63. Energy and velocity being so nearly in true similarity as is there shown, area will follow closely the trend of specific volume. Examination will show that the differences among the MN curves in Fig. 64 correspond almost exactly with those among the true adiabatics in Fig. 50, due account being taken of the different base scales in the two diagrams. Down to $p/p_1 = 0.4$, the jets conform within 0.25 per cent; but at the lower limit of 1 lb. absolute pressure (indicated by cross marks on the curves) the disagreement is from 1.0 to 1.7 per cent.

(*h*) THE NAPIER DIVISOR. — In order to fill out the columns for R_a and a_0 in Table 7, it is best to interpolate first the Napier divisor

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D. Napier's formula, long in use to calculate the flow of steam through an orifice, is

$$f = \frac{ap}{70}; \quad \ldots \ldots \ldots \quad (106)$$

in which a is area of orifice in square inches, p is initial absolute pressure (our p_1), and f is the weight of steam discharged per second. This formula holds if the low-side or terminal pressure p_2 is not greater than $0.6\ p_1$ — it may have any lower value That the pressure p_2, provided only that it is below the throat pressure, has no appreciable effect upon the rate of flow is an important fact; it implies that the steam jet so

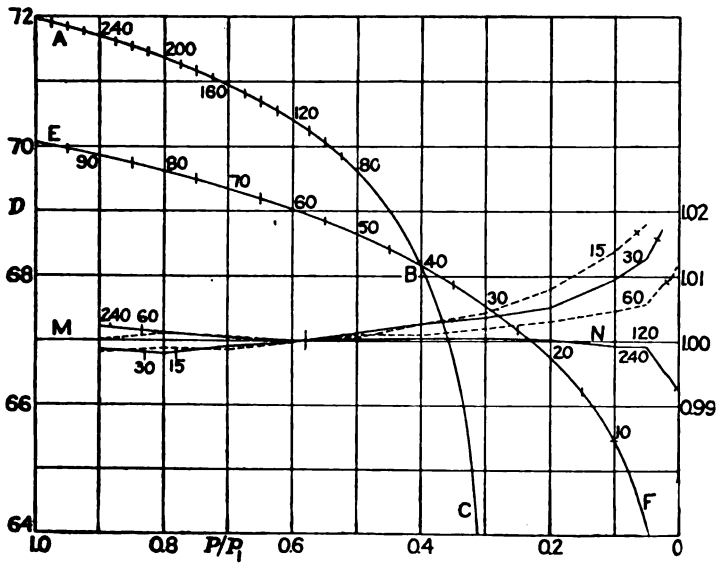

FIG. 64. — Comparison of Jet Areas.

Similar to Fig. 63 in general scheme. The curves on base MN, to scale at right, show departure of jets from true similarity: see § 16 (*g*). The curve ABC–EBF is the Napier divisor D, to the scale at the left, and with the scale of determining, initial pressure which is marked along the curve; see § 16 (*h*).

adjusts itself as always to locate the throat of the jet in the smallest part of the orifice or nozzle. Whether or not the enlarging nozzle is present, to control expansion beyond the throat pressure, is a matter of very minor influence as regards rate of discharge. Further discussion of this question will be found in Chapter IX.

Now putting our a_0 for the a in Eq. (106), considering the unit jet so that $f = 1$, and using a general symbol D in place of the divisor 70, we get

$$1 = \frac{a_0 p_1}{D}, \qquad D = a_0 p_1. \quad \ldots \ldots \quad (107)$$

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The variation of this D with p_1, for the ideal unit jet, is represented by the curve ABC–EBF in Fig. 64. This curve was laid out with a number of plotted points, more than those for the five jets in Fig. 62, and the tabular values were measured from it. Having D, division by p_1 gives a_0, col. 5 of Table 7; and division of these numbers by the particular a_0 for 120 lb., or by 0.5868, gives R_a, which is the ratio of the absolute sizes of the jets. Finally, the degree of variation in the divisor D is shown by the ratio R_D, which is $(D \div D_{120})$ Over the range of common boiler pressures D keeps close to 70, but for low initial pressures it falls off quite rapidly.

EXAMPLE 19. —With initial dry steam at 100 lb. absolute, compute the ideal flow through an orifice 0.5 in. in diameter, first, when the low-side pressure is 80 lb. absolute; second, when it is 15 lb. absolute.

The area of the orifice is 0.1964 sq. in., and we assume that it has a properly rounded approach or entrance.

For 100 lb., throat area a_0 is 0.7006 sq. in., by Table 7. For the pressure ratio 0.8, a/a_0 is 1.166 by Table 6. Then the unit area in the first case is

$$a = 0.7006 \times 1.166 = 0.8169 \text{ sq. in.}$$

In general, the ideal rate of flow is equal to area of orifice divided by least area of unit jet. With discharge into 80 lb. pressure, the least area is that just computed, so that the rate of flow is

$$f = 0.1964 \div 0.8169 = 0.2405 \text{ lb. per sec.}$$

With 15 lb., or with any other discharge pressure less than $0.58\,p_1$, we use the throat area a_0; then the flow in the second case is

$$f = 0.1964 \div 0.7006 = 0.2803 \text{ lb. per sec.}$$

(*i*) THE EFFECT OF INITIAL CONDITION upon the form and dimensions of the ideal steam jet must now be considered. The comparison made in Fig. 65 covers this matter with sufficient completeness. For an initial pressure of 120 lb. absolute, the same as the "standard" in Tables 6 and 7, jets have been computed for $x_1 = 0.9$, 0.8, 0.7, and 0.0, and for initial superheats of 100, 200, 300, and 400 deg. The jet areas are laid out directly in the group of curves marked A, after the manner of Fig. 62, with the initial dry-steam curve emphasized. The three jets for high values of x_1 are very similar to that for $x_1 = 1.0$, growing smaller in area with x_1 at a slowly increasing rate; but the curve for all water at the start, or for $x_1 = 0.0$, differs in having the throat or minimum area come much earlier on the pressure base, and then showing a much higher rate of enlargement. The only striking feature of the superheat curves is the general tendency to become relatively smaller, or to have a smaller rate of enlargement after passing the throat;

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this is shown especially by their crowding against the dry-steam * curve at the low end of the pressure range.

A much clearer showing of the relative sizes of the several jets is made in the B curves. At any pressure ratio, or on any ordinate line,

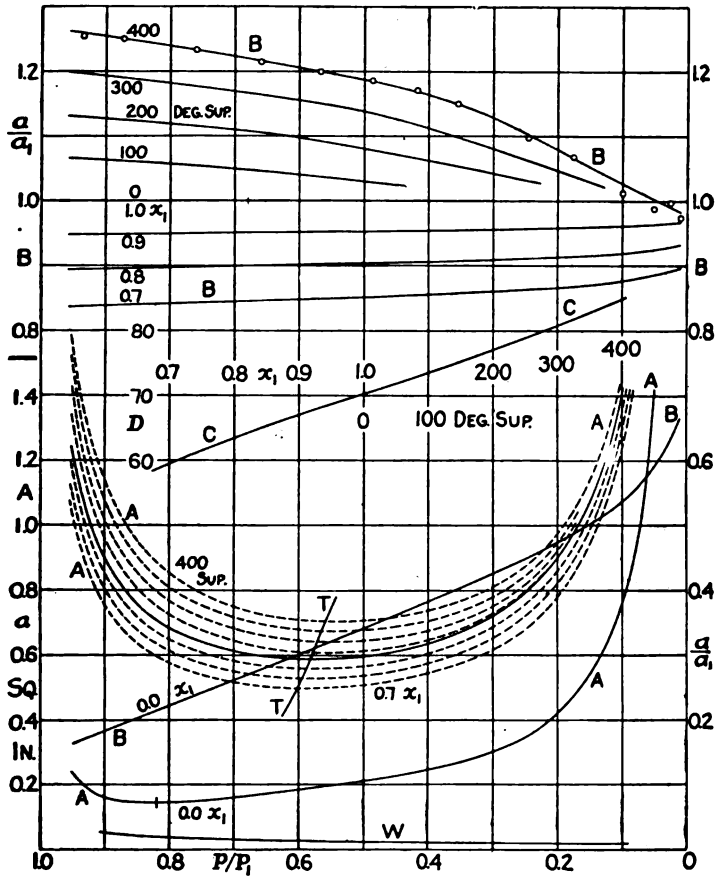

FIG. 65. — Comparison of Jets with Different Initial Condition, All for 120 Lb. Initial Pressure.

Curve groups A and B are on the base of pressure ratio; group A shows jet areas directly, group B relative areas. Curve C, on a base of quality x and superheat s, gives the Napier divisor D.

the area $a_{1.0}$ of the dry-steam curve is taken as base, and the quantity plotted is the ratio of the areas of the other jets to this standard. Here, then, we compare actual areas, as against the comparison of relative areas in Fig. 64, where each a was first referred to its own a_0, so as to

* The word initial is understood; its omission, here and later in the discussion, need cause no confusion.

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give the curves a common point at the throat of the jet. As an example of the present method, at $p = 0.8\,p_1$ the area of the dry-steam jet is 0.684 sq. in., from data in Table 6; the corresponding area of the 0.7 jet is 0.577 sq. in.; and the ratio $0.577 \div 0.684 = 0.844$ is the ordinate of a point in the B curve for $x_1 = 0.7$.

It is now apparent that the wet-steam curves have a slight tendency to approach the "standard" as the pressure falls, or to become of the same absolute size, which tendency increases as x_1 is smaller, and becomes stronger in the region of very low pressure. The superheat curves show the same tendency in much greater degree, and at low pressures they run into close agreement with the (initial) saturation curve, the line MN. The determination of the superheat curves, with data from Tables VI and VII, is necessarily less precise than the pure computations for adiabatics all in the region of saturated steam; consequently, with a possible error of perhaps one per cent in the "points" for these curves, too close inferences must not be drawn from the exact form of the B curves as here laid out.

(*j*) THE PRINCIPAL INFLUENCE affecting the form of the jet, as measured by the instantaneous area a, is the varying specific volume v. Referring to Fig. 61, we note that if the expansion curve BC is somewhat modified in form, so that v increases at a greater or less rate than in that diagram, there will be a slowly accumulating change in the value of area ABCD or energy E; but this change in E will be much less in proportion than the change in final v. The effect of variant initial condition upon the volume in adiabatic expansion has been brought out in Fig. 51, and it will now be made clearer by a numerical comparison. At a final pressure $p_2 = 0.1\,p_1$, the ratio of final volume v_2 to initial volume v_1, for each of the eight closely grouped curves in Fig. 65 is as follows:

	x_1				s_1			
	0.7	0.8	0.9	1.0	100	200	300	400
v_2/v_1	8.08	7.88	7.73	7.60	6.87	6.30	5.89	5.70

Here s is used as a symbol for superheat in degrees, not to be confused with specific saturation volume. For the one case not in this tabulation, that of $x_1 = 0.0$, the corresponding ratio is about 240.

A basis for the comparison of energy development is derived from the idea, made obvious by Fig. 61, that energy E is some function of the original product p_1v_1; then a ratio of E to p_1v_1, or, in the present case of constant p_1, of E to v_1, is a good measure of relative effect. One example will suffice: with $p_1 = 120$ and $x_1 = 1.0$, v_1 is 3.728 cu. ft.; and after expansion to 12 lb., E is 165.9 B.t.u. by Table 6; the ratio of

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these numbers is 165.9 ÷ 3.728 = 44.50. With $x_1 = 0.7$, v_1 is 2.615, E is 120.1, and the ratio is 120.1 ÷ 2.615 = 45.82. Now this ratio 45.82 is only 3 per cent greater than 44.50, while the volume ratio 8.08 is 6.3 per cent greater than 7.60, above. Since velocity varies as the square root of energy, so that the percentage of difference will be reduced to 1.5 in this case, it appears that in a drop of x_1 from 1.0 to 0.7 there is a relative increase in volume more than four times as great as the similar increase in velocity, and this accounts for the upward slant of the B curve for 0.7 in Fig. 65. With initial superheat, both volume and energy (referred to initial volume) fall off more rapidly than with saturated steam; but here again volume changes much the faster, hence the relative decrease in area of jet.

(*k*) Flow of Steam. — To put this matter into definite shape for the particular initial pressure in Fig. 65, the least area a_0 (which varies in position on the base scale according to the approximate locus TT) is measured and multiplied by p_1 or 120, after the manner of Eq. (107). The resulting values of D are plotted in the curve CC, and from the smoothed curve Table 8 is obtained. In the second part of the table, the determinant is not degrees of initial superheat, as in the diagram, but the ratio of this superheat s or $(t - t_s)$ to the absolute temperature $T_s = t_s + 460$ at saturation.

Table 8. Napier Divisor D, for $p_1 = 120$ Lb. absolute, Varying with Initial Quality or Condition of the Steam in the Jet.

			For Wet Steam				
Quality X_1 =	0.7	0.75	0.8	0.85	0.9	0.95	1.0
D =	59.6	61.6	63.5	65.3	67.1	68.8	70.4
			For Superheated Steam				
Ratio of superheat =	0.0	0.1	0.2	0.3	0.4	0.5	0.6
D =	70.4	73.0	75.7	78.5	81.5	84.6	87.9

Now without undertaking the laborious task of making a number of comparisons like Fig. 65, we can get a working method for calculating flow of steam by assuming that the D curve for other initial pressures will be like that in Fig. 65, as set forth in Table 8. Then the ratio R_D in Table 7 can be used for adjusting values of D other than those belonging to the case of initial dry saturated steam. And even if the discharge pressure is higher than the throat pressure, at least a roughly approximate result can be got with the help of the ratio a/a_0 in the upper part of Table 6. The use of these methods, and the degree of error in their results, can best be shown by an example.

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EXAMPLE 20. — Find the ideal rate of flow per square inch of orifice in the following cases:

(*a*) Pressure 60 lb. abs., quality 0.75, flow into atmosphere.

(*b*) Pressure 200 lb. abs., superheat 320 deg., flow into 110 and into 160 lb.

Case (*a*). Take from Table 8 the value $D = 61.6$, and to change from 120 lb. to 60 lb. multiply it by $R_D = 0.980$ from Table 7; then the new value of D will be $61.6 \times 0.980 = 60.4$. Dividing this by p_1 or 60, according to Eq. (107) we get $a_0 = 1.006$; whence the flow per square inch is $1 \div 1.006 = 0.994$ lb. per second.

Case (*b*). At 200 lb. pressure the saturation temperature is 382 deg. fahr. or 842 deg. absolute; then the "ratio of superheat" is $320 \div 842 = 0.380$, for which D_{120} is 80.9 by Table 8. The factor $R_D = 1.0135$ from Table 7 changes this to $D = 80.9 \times 1.0135 = 82.0$ for the jet from 200 lb. Division by 200 gives $a_0 = 0.410$, which is the least area when the discharge pressure is 110 lb.; and for that condition the flow per square inch will be $1 \div 0.410 = 2.44$ lb. per sec.

With $p_2 = 160$ lb. the pressure ratio is 0.8, and a/a_0 from Table 6 is 1.166; this changes a to $1.166 \times 0.410 = 0.478$, and the rate of flow becomes $1 \div 0.478 = 2.09$ lb. per sec.

To check the results in case (*b*), calculation by the "exact" method will now be made. At 200 lb. pressure and 320 deg. superheat, the temperature t is 702 deg.; under these conditions the total heat h_1 is 1370.8 B.t.u. by Table VII, and the entropy n_1 is 1.721 by Table VIII. Running down this line of constant entropy we read $t = 643$ deg. at 160 lb. and $t = 548$ deg. at 110 lb. pressure, in Table VIII. For these pressures and temperatures, h_2 is read as 1344.0 and 1300.5 B.t.u. The derived results are as follows:

	At 160 lb. discharge.	At 110 lb. discharge.	
Energy $E =$	26.8	70.3	B.t.u.
Velocity $V =$	1155	1870	ft. per sec.
Specific volume, by Table VI, $v =$	4.07	5.38	cu. ft.
Area $a =$	0.507	0.415	sq. in.
By approximate method above, $a =$	0.478	0.410	sq. in.
Per cent of error =	5.7	1.2	per cent.

It appears, then, that for a discharge determined by throat area the "approximate" method is quite good; but for a higher discharge pressure it is so rough as to be of little use. The last conclusion is in full accord with the form of the B curves for high superheat in Fig. 65.

(*l*) FLOW OF WATER. — There is positively no use in carrying the comparison in Fig. 65 below $x_1 = 0.7$, so far as steam and water mixtures are concerned. But the extreme case of no steam at the start, or of the efflux of hot water, is of interest as underlying the actual flow of water through a boiler blow-off or try cock. Beneath the A curve for $x_1 =$ zero, in Fig. 65, is drawn the curve W, which shows to the same scale the ever-decreasing area of an ideal jet of cold water, driven

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by the initial pressure 120 lb. The important facts brought out by the curve for hot water are: first, that only a small range of pressure drop (down to about $0.82p_1$ in the figure) is available to drive the jet through the orifice; second, that even at this high throat pressure, the area is nearly four times that of the cold-water jet; in exact terms, this example shows the maximum ideal flow of hot water from 120 lb. as but 0.27 of the flow of cold water from 120 lb. to 98 lb. pressure. This marked effect of volume increase, due to the formation of steam within the jet, in cutting down the rate of discharge accounts for the lightness of the jet from a boiler blow-off, as compared with a jet of cold water from an equal orifice and driven by the same pressure.

(*m*) Actual Flow of Steam. — The whole purpose of this section has been to put the ideal jet into shape for ready quantitative use. The essential requirement of the perfect jet is that all the convertible energy shall be put into the form of kinetic energy of the steam current. In the actual jet, this transformation fails of completeness in several directions. The closer study of these wastes is taken up in Chapter IX, as a preliminary to the analysis of steam action in the turbine; but a brief general description of them is appropriate here.

With actual material for nozzle and other confining surfaces, there will be some loss of heat by conduction and radiation. Also, there is some tendency to abstract heat from the hot, high-pressure steam, conduct it along the nozzle, and give it back to the cool, low-pressure steam; but the possibility of such transfer action is absolutely insignificant when compared with its degree of prevalence in the piston engine.

The nozzle surfaces offer a resistance to steam flow, of the general nature of friction. Mechanical energy thus absorbed goes back into the steam as heat, but under the penalty of only a small possibility of subsequent reconversion.

Mechanical energy used to produce motions other than the direct forward flow of the current is wasted; anything that causes churning or whirling motion within the jet acts in this direction. Nonhomogeneity of the substance, especially if it take the form of drops of water mixed with the steam at the start, greatly hinders proper jet formation; a drop of water will receive a much smaller acceleration, by a given unbalanced unit pressure, than will the same mass of steam, and each drop will serve as a center of eddying motion. All energy of secondary motion finally settles into heat in the steam at low temperature.

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§ 17. Throttling or Kinetic Pressure Lowering

(*a*) THE ACTION OF THROTTLING. — This name is given to the process through which the pressure of a current of steam is reduced from a higher value p_1 to a lower value p_2, by free or unresisted expansion. It occurs whenever steam flows through a partly opened valve or some other restricted portion of its channel, so that pressure work is used up in producing velocity and kinetic energy, but without any useful application of this energy. The term "wire drawing" is sometimes used as a synonym for throttling.

This operation is so closely connected with jet formation, that we shall take the latter as a point of departure. Suppose steam to flow

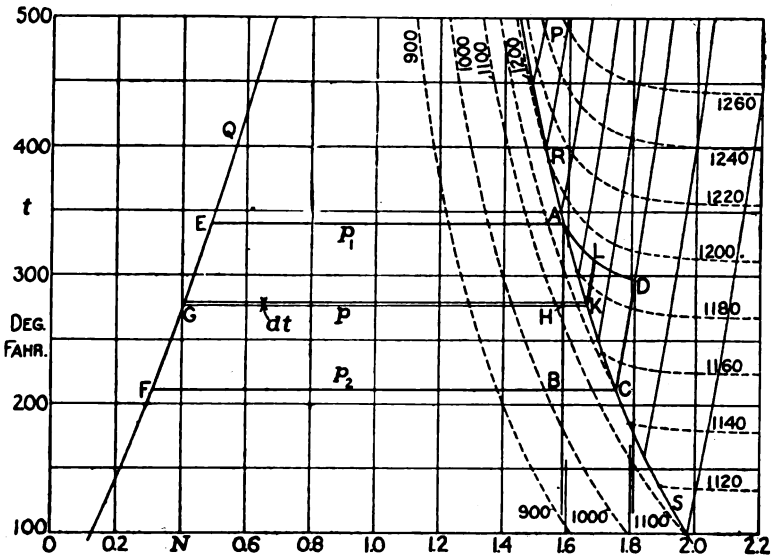

FIG. 66. — Temperature-entropy Diagram, to Show Throttling Effect and Lines of Constant Total Heat.

through an orifice or nozzle from pressure p_1 to pressure p_2. If the jet is perfectly formed, its kinetic energy will be equivalent to the area EABF on the temperature-entropy diagram, Fig. 66; the condition for "perfection" is truly adiabatic expansion from A to B. Now instead of using the steam jet to drive a turbine wheel or for some other effective purpose, let it simply come to rest in the low-pressure chamber, the kinetic energy being dissipated in whirls and eddies. The ultimate result is the reconversion of all the mechanical energy into heat; and as developed this heat comes back into the steam at p_2 until (if there has been no loss by radiation) the total heat has the same value as at

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A. In Fig. 66, the heating at constant pressure p_2 is represented by BC out to the saturation line, then by CD in the region of superheat. The area under BCD, down to absolute zero, must equal area EABF. The point D lies on a line of constant total heat AD, drawn from A.

(*b*) CONSTANT TOTAL HEAT. — As already defined in several places, the total heat of steam is the amount of heat required to change water at 32 deg. into steam at a condition made definite by p, v, and t, the whole operation being carried out under the constant pressure p. For this general value the symbol h has been used, and it is the sum of the internal or intrinsic energy and of the external work or energy APv. In Fig. 66 the dotted curves are lines of constant total heat. For wet steam they are found from data in Table II, for superheated steam readings must be taken from Table VII.

As regards the shape of these curves, it will be noted that in the region of wet steam they have the general trend of the saturation line, running nearly "parallel" to it at the top of the diagram, where the total heat H of dry steam is nearly constant, but bending toward it at an increasing rate as the pressure falls. With superheated steam, the curves show a rapidly dropping temperature at high pressure, but become nearly horizontal (isothermal) as the pressure approaches zero. This meets rational requirements, since steam approximates a perfect gas in behavior when pressure and density are low, and for such a gas the isothermal is the line of constant total heat.

EXAMPLE 21. — For the pressure 246.9 lb. (at 400 deg. saturation temperature), determine points on the total heat curves for 1100 B.t.u. and 1260 B.t.u.

From Table II, at 400 deg., $H = 1201.9$, $r = 827.9$; the total heat $h = 1100$ is 101.9 B.t.u. short of complete evaporation, or $m = 101.9 \div 827.9 = 0.1230$. The entropy b, length QR in Fig. 66, is 0.9631; then the distance from R to the 1100 B.t.u. line is $0.1230 \times 0.9631 = 0.1185$ of entropy.

For the other point sought, refer to Table VII. Run up the 1260 B.t.u. line to 246.9 lb. On this pressure line (horizontal), the distance between the 490 deg. and 500 deg. isothermals is 5.5. B.t.u., while the 1260 line is 1.8 B.t.u. out from the 490 deg. line. To 490 deg. add $10 \times 1.8 \div 5.5$ or 3.3, getting 493.3 deg. as the temperature corresponding to 1260 B.t.u. By measuring this temperature up along the pressure line PR, the point where the constant-heat line crosses that curve is fixed.

(*c*) CONTINUOUS THROTTLING. — In the operation represented by the outline AB–BCD, Fig. 66, jet formation or steam acceleration is distinct from and antecedent to the dissipation of kinetic energy, and the curve AD is merely a line of relation. This is an extreme and limiting case, never quite realized because perfect jet formation is impossible

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under actual conditions. At the other extreme lies the operation of continuous or gradual throttling, which would be represented by the constant-heat curve AD. This is realized when steam is passed through a porous plug, or when a current of steam gradually falls in pressure in flowing along a nonconducting pipe. As fast as available energy is converted into the mechanical, kinetic form, it is changed back into heat through the overcoming of what we may call frictional resistances. If the process is truly continuous, the total heat *in* the steam will remain constant, and the drop from p_1 to p_2 will take place along the temperature-entropy line AD.

Throttling, in the steam-power plant, usually occurs in such fashion that it would be represented by some curve intermediate between ABCD and AD: but no matter what is the detail of the process, the net effect is a change of state from point A to point D on the same total-heat line.

(*d*) INCREASE OF ENTROPY IN THROTTLING. — In the operation of throttling, whether continuous or by parts, there is increase of entropy without reception of heat from any external source. At first sight, the process is adiabatic, and yet it is not isentropic. Any mental confusion over the matter will disappear, however, when it is remembered that the entropy held by a body is something which changes only when energy goes into or out of the substance in thermal form. In adiabatic expansion heat is changed into mechanical work, and its exit (from the store of heat) in this form does not affect the content of entropy; but when, in throttling, the mechanical energy of the steam current is changed back into heat and added to the store of heat, it brings entropy with it, just as if it were heat added from without to a quiescent body of steam. The process is adiabatic as regards the surroundings, but is not adiabatic as regards the steam itself.

(*e*) ENERGY TRANSFORMED IN THROTTLING. — In Fig. 66, area EABF represents the portion of the total heat (at A) that is changed into kinetic energy, and area under BCD shows the same energy as changed back to heat in the steam. For the case of continuous throttling, area EADCF, equal to area under AD, is the energy which goes through this double transformation. The difference is due to the fact that in the latter case heat is being continually returned to the steam, instead of being kept in the form of mechanical energy till the lower pressure is reached. The element of jet energy $v\,dp$ in Fig. 61 becomes, in Fig. 66, a narrow strip GH or GKL, which can be expressed as $n\,dt$ (with due regard to the proper method of measuring the n factor): then with complete jet formation the energy given to the jet at p, and kept in it down to B, is the area of strip GH; in porous plug throttling the energy given and immediately returned is strip GKL.

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Fig. 67 shows the pressure-volume diagram corresponding to the part of Fig. 66 which illustrates our example of throttling. Development of the adiabatic jet, followed by dissipation of velocity at the lower pressure, is represented by the outline AB - BD, continuous throttling by the simple curve AD. The larger specific volume gained by keeping the total heat up to its full value, in the second case, is clearly shown, with the resulting increase of $v\,dp$ from strip GH to strip GL.

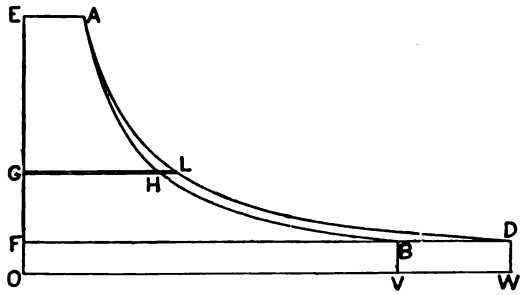

Fig. 67. — Pressure-volume Diagram for the Two Cases of Throttling Action.

In final statement, the area ABCD in Fig. 66 or ABD in Fig. 67 is accounted for by saying that it represents energy which goes *twice* through the cyclical transformation from heat to work and back again, while EABF follows this cycle but once.

(*f*) Reversibility of the Steam-jet Cycle. — In respect to the operations performed within the engine, the Rankine cycle, diagrammed in Figs. 55, 56, 61, and 66, is reversible. This is true whether work is done upon a piston or in accelerating the steam jet, provided only that the expansion with fall of pressure from p_1 to p_2 is isentropic. A swiftly flowing current of gas or liquid can, in a suitably formed retarding tube, be brought to rest with rise of pressure, and with the transformation of a large proportion of its kinetic energy into static-pressure work against the higher pressure. Ideally, if there were no dissipation of kinetic energy in secondary motions or in overcoming frictional resistances, the perfect steam jet would be completely reversible, and the steam could be driven right back into the chamber where the pressure p_1 exists.

Examples of the reversed pressure-to-velocity cycle are found in the steam-jet blower and the centrifugal fan, the injector and the centrifugal pump. Complete reversibility, conditioned by equivalence between the kinetic energy of a jet and the static-pressure work which it can do in being brought to rest, is far from attainable. In general, the velocity-to-pressure cycle has a decidedly lower efficiency, actual and relative, than the direct or velocity-producing operation, accounted for by the

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greater tendency to waste motion, with the ultimate conversion of useful energy into heat at low temperature. The same ideal of complete conversion of available energy underlies both actions, but in one direction the unavoidable wastes are greater than in the other.

(*g*) HEAT WASTE IN THE STEAM-JET CYCLE. — The present chapter goes no farther than the formation of the jet, leaving for later discussion all questions as to the manner of harnessing it for useful output. In whatever way the jet is used, however, there will be some loss of ideal effect in its formation, and a further (and larger) loss in its application. The net result will be an increase of entropy from source to receiver, like that from N_1 to N_2 in Fig. 38 and from N to T in Fig. 58. Full throttling or complete dissipation of available energy, with reduction of the steam to the final state at D, Fig. 66, is an example of zero efficiency, of the same class as that in Fig. 39. In any working apparatus, the energy waste will be of the order of magnitude shown in Fig. 58.

It may be well to restate the fundamental fact that the rejection of the heat under line NE in Fig. 58 or line BF in Fig. 66 is absolutely unavoidable. The further rejection represented by area under TN in Fig. 58 is due to causes which, while they can never be eliminated, yet have their activity largely determined by the design and the manner of operation of the machine.

(*h*) FALL OF TEMPERATURE IN THROTTLING. — This has been shown in Fig. 66, and that it must always take place in the throttling of superheated steam is indicated by the relation between the curves of constant total heat and of constant temperature in Table VII. The exact manner of this temperature change is shown in Fig. 68, in a direct plot of temperature on pressure, for which points are got from Table VII as in the latter part of Example 21. There are two groups of the constant heat curves HH. Those which have their origin (at the upper end) on the saturation line SS are determined by the initial pressure of saturation with which they are marked, and the value of the total heat along each can be found by reference to Table II. The upper curves do not touch the saturation line within the limits of this diagram (and above 1210 B.t.u. are entirely clear of it): they are drawn for every 5 B.t.u., as indicated.

In connection with the lower set of curves in Fig. 68 are plotted three important sets of experiments upon the change of temperature in throttling. For any row of points there is a fixed initial pressure p_1, and each point represents the temperature observed in an experiment made with a particular p_2. The purpose in plotting the points here is to compare their trend with the curves derived from Table VII, which itself is largely based upon these experiments. The more irregular points are

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strung together by broken lines, merely to show that they belong to the same series. The apparatus used by the several experimenters, differing

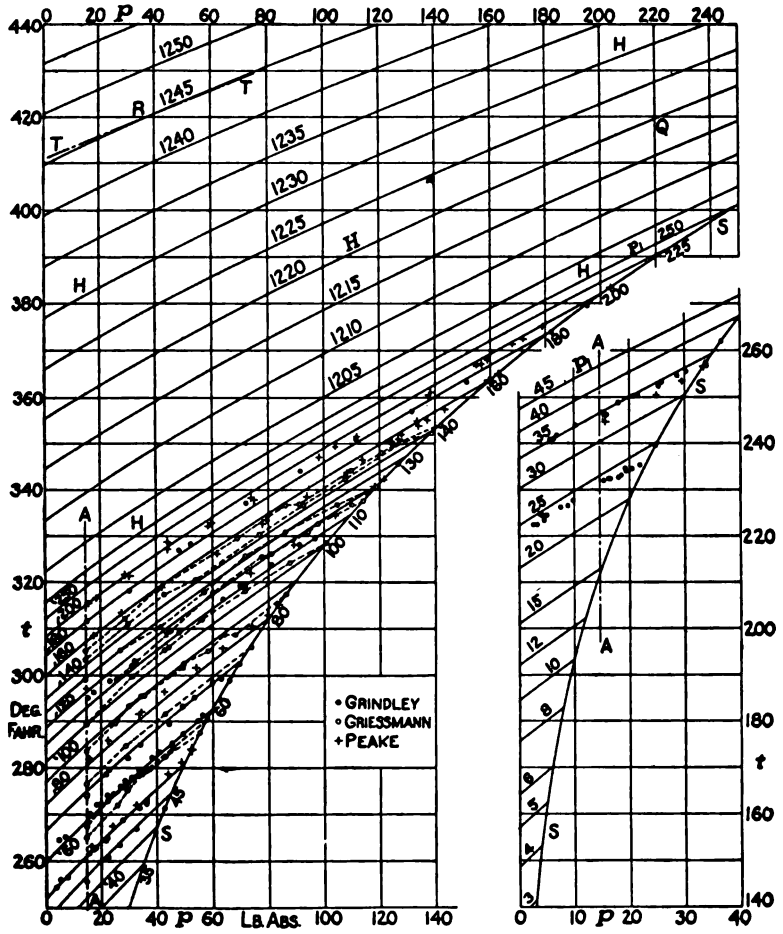

FIG. 68. — Fall of Temperature in Throttling, or Curves of Constant Total Heat in a Plot of Temperature t on Pressure p.

For references concerning the throttling experiments plotted, see Note 7, page 617, Appendix.

quite widely in detail, was in every case a throttling calorimeter, made and operated with especial care.

It is an important proposition in the physics of gases, that in constant-heat pressure lowering the rate of change of p with t is wholly a function of t; in other words, that curves like those in Fig. 68 should all have the same inclination at any particular horizontal line of equal

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temperature. For the purpose of testing the conformity of superheated steam to this law, comparison is made between points Q and R in the figure. The line TT is drawn parallel to a tangent at Q on the curve for 1220 B.t.u.; and instead of being tangent at R is found to make a small angle with the curve for 1245 B.t.u. This discrepancy is due either to an as yet insufficiently accurate determination of the properties of steam, or possibly to a slight departure from the law in the region near to saturation.

(*i*) THE THROTTLING CALORIMETER. — This instrument, based upon the fact that in throttling or free expansion there is no change of total heat (unless through escape in radiation), is used to find the quality of moist steam. As outlined in Fig. 69, steam is drawn from the steam

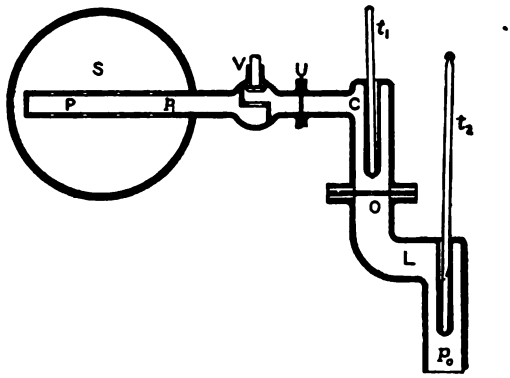

FIG. 69. — Throttling Calorimeter.

main S, through the perforated pipe P and open valve V, into the chamber C where, at pressure p_1, the temperature t_1 is read. It then flows through the small orifice O into the low-pressure chamber L, open to the air, and the temperature after throttling is read as t_2. Provided only that this t_2 is greater than t_{s2} (here 212 deg.), the total heat is definite in terms of pressure and temperature; and being measured at p_2 is known in value for the steam as originally at p_1. The actual total heat h_2 (equal to h_1) can be read from Table VII for p_2 and t_2; then for the original deficiency in total heat due to incomplete evaporation, and for the fraction of initial moisture we have,

$$m_1 r_1 = H_1 - h_2, \qquad m_1 = \frac{H_1 - h_2}{r_1}. \quad . \quad . \quad . \quad (108)$$

EXAMPLE 22. — In a throttling calorimeter which discharges into the atmosphere, the temperature readings (corrected) are $t_1 = 334.5$ deg., $t_2 = 261.4$ deg. What is the quality of the steam tested, assuming no radiation from the calorimeter?

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The pressure corresponding to 334.5 deg. is 109.5 lb. absolute or 94.8 lb. by gage, the total heat of dry steam is $H_1 = 1188.1$ B.t.u., and the latent heat r_1 is 883.0 B.t.u. At the pressure of the atmosphere and at 261.4 deg. the total heat h_2, by interpolation between the isothermals for 260 and 270 deg. in Table VII, is 1173.4 B.t.u. This is also the actual total heat of steam at p_1 with moisture m_1; then the heat shortage is $m_1 r_1 = 1188.1 - 1173.4 = 14.7$ B.t.u., and the fraction of moisture is

$$m_1 = 14.7 \div 883.0 = 0.0167, \text{ or } 1.67 \text{ per cent.}$$

(*j*) DIAGRAM FOR THE THROTTLING CALORIMETER. — To facilitate the calculation just illustrated, several important quantities are diagrammed in Fig. 70, for the case of discharge into the atmosphere. The

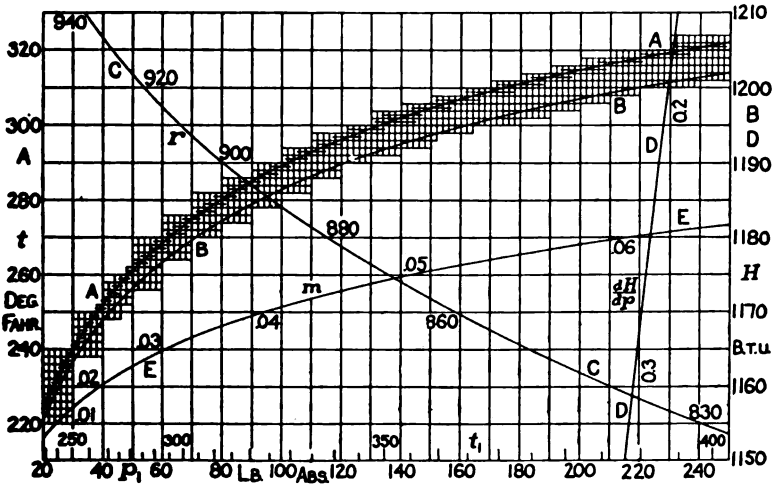

FIG. 70. — Diagram for the Throttling Calorimeter.

base is initial pressure p_1, with which is laid out a variable scale of corresponding saturation temperature t_1. Curve AA shows, to the scale at the left, the temperature t_2 which results from the throttling of initially dry steam to atmospheric pressure, or to $p_2 = 14.7$ lb. absolute; note that while this is *for* p_1 as scaled below, it is *at* p_2. Curve BB is the total heat H_1 of initially dry steam, existing also at p_2 as the value of h_2 correlative with the maximum t_2 given by curve AA; in the latter relation, h_2 on BB is directly under the corresponding t_2 on AA. To obviate the need of looking up r_1 in Table II, it is laid out in curve CC, from which it can be read with sufficient accuracy. If the existing pressure of the atmosphere is something other than 14.7 lb. or 29.92 in. of mercury, a correction must be made in the value of h_2 for t_2. Reference to Table-diagram VII shows that for a given temperature t the total heat h increases as pressure falls, decreases as it rises, so that the

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rate of change is negative. The coefficient of variation, dh/dp (which changes with the slant of the isothermal lines in Table VII), is given by curve DD, for which the base is the scale of B.t.u. at the right; the rate is expressed in B.t.u. per pound of pressure. Finally, curve EE shows, for any p_1, the limiting or maximum value of moisture m_1 that can be measured by the calorimeter with discharge into the atmosphere; this matter is explained in the next article.

EXAMPLE 23. — By means of Fig. 70, find indicated moisture from the following data: Temperatures, $t_1 = 342.6$ deg., $t_2 = 242.7$ deg.; barometer reading 29.3 in. mercury.

For $t_1 = 342.6$, p_1 reads about 122 lb. by the parallel scale of pressures (122.1 by table), and the total heat of dry steam, from curve BB above this pressure, is $1190.1 = H_1$. With dry steam, from this pressure, the temperature after throttling would be 297.0 deg., as read from curve AA at the ordinate for $p_1 = 122$. The difference between this and the actual $t_2 = 242.7$ is due to the initial wetness of the steam tested.

Now locate the temperature 242.7 deg. on curve AA (as a vertical measurement, according to the scale at the left), and read beneath it on curve BB the value 1164.6 B.t.u. for h_2. This must be corrected for the barometer reading; and since the latter is below normal, h_2 must be increased. The pressure p_2 is 0.62 in. or 0.3 lb. low; from curve DD the coefficient of variation, at $H = 1165$ is 0.31 B.t.u. per pound, so that h_2 must be increased by $0.3 \times 0.31 = 0.09$ or 0.1 B.t.u., making it 1164.7. The heat shortage or moisture effect is now

$$m_1 r_1 = H_1 - h_2 = 1190.1 - 1164.7 = 25.4 \text{ B.t.u.}$$

From curve CC, $r_1 = 877$, then $m_1 = 25.4 \div 877 = 0.0290$, or 2.9 per cent.

In the use of this diagram, there is liability to an error of about 0.5 B.t.u. in $m_1 r_1$, from purely graphical inaccuracy; but this is a good deal less than the possible experimental error.

(*k*) RANGE OF THE CALORIMETER. — As suggested in the example just worked, the effect of initial moisture is to lower the temperature t_2, diminishing the superheat of the discharged steam. Evidently, if the wetness at p_1 exceeds a certain amount, t_2 will fall to 212 deg. (or to the saturation temperature at any p_2), and the final state of the steam will cease to be determinate with respect to total heat. Curve EE in Fig. 70 shows the maximum measurable value of m_1. To find this amount of moisture which will just lower t_2 to 212 deg., subtract from H_1 the total heat $H = 1149.7$ at 212 deg., and divide this heat difference by r_1. If, for instance, the existing total heat at 200 lb. pressure is 1149.7, the full heat being 1198.6, the shortage is 48.9 B.t.u.; and with the latent heat $r_1 = 843.8$, the fraction of moisture is 0.0580. The last number gives a point on curve EE.

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In practice, it is not well to use the throttling calorimeter alone if t_2 falls below 220 deg., with atmospheric discharge.

(*l*) THE SEPARATOR CALORIMETER. — With steam that is quite wet, a small separator, similar to those used in pipe lines for the purpose of removing water from the steam current, is a very effective quality meter. As outlined in Fig. 71, the separator is intended to be coupled in on the high-pressure side of the throttler in Fig. 69, at the union U, when the steam is too wet for the first instrument. This combination constitutes the older form of the Barrus Universal Calorimeter; the separator takes out nearly all of the moisture from the sample, sending almost dry steam over into the throttler. If the separator is to be used alone, there must be a throttle valve or a discharge orifice in its steam outlet. In any case, the drain valve D is so manipulated as to keep the water in the chamber C at or near a certain level (shown in the glass gage G), well below the mouth of the inlet pipe P.

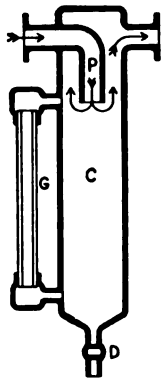

FIG. 71. — Separator Calorimeter.

If g_1 = weight of steam discharged in a certain time and g_2 = weight of water separated, so that $g = g_1 + g_2$ is the total weight of mixture subject to observation, the moisture indicated is, very simply,

$$m_1 = \frac{g_2}{g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (109)$$

(*m*) ACCURACY OF STEAM CALORIMETRY. — Along several lines error and unreliability may creep into the determination of quality of steam by the methods just given. These will now be briefly described, but not discussed in detail.

The sample drawn into the calorimeter is likely to be far from representative of the current in the steam pipe. Moisture in excess of a very small proportion by weight tends to settle out or segregate, and the resulting nonuniformity will be greater when the pipe is horizontal, as the flow is slower, and when the steam is not stirred up by obstacles and changes of direction. It is best to place the sampling pipe in a vertical part of the steam line, and just beyond a bend; but in many cases the only sure thing is to have a separator in the line, weigh its discharge of water from the whole current, and use a calorimeter to test the steam flowing from it.

Loss of heat by radiation from the instrument will cause the indicated moisture fraction to be greater than that really existing in the steam tested; it lowers t_2 and h_2 in the throttling calorimeter, increases

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g_2 in the separator. Radiation is to be prevented as far as possible by nonconducting covering, and against what is unavoidable the instrument must be calibrated, by passing through it steam known to be dry; the results of the calibration are introduced as corrections in calculating from ordinary readings.

Errors in thermometer indications, the greatest due to partial immersion in shallow thermometer cups, must be determined and allowed for. The measurement of temperature should be essentially correct, although extreme precision is not called for.

There is considerable possibility that the steam in the low-pressure chamber of the throttling calorimeter may not be homogeneous, especially if drops of water come in as a part of the sample. Experiments upon the heat required for superheating steam have shown, incidentally, that a small proportion of moisture may exist in a current of much hotter superheated steam for an appreciable time; and this condition would persist much longer with comparatively large masses of liquid than with the extremely fine mist produced by condensation in adiabatic expansion. The effect of this action will be to raise t_2 and h_2 above their true values, and its possibility is a strong argument against using the throttler alone with more than two or three per cent of moisture.

The more detailed consideration of these matters, as also the description of other schemes, belongs to a treatise on laboratory and experimental methods.

§ 18. Special Graphical Methods

(*a*) THE IDEAL STEAM CYCLE, whether for engine or turbine, takes the form of the Rankine cycle, Figs. 55, 56, 61, and 66. The quantities involved in calculations upon this cycle are set forth, in various relations, in Tables II, VI, VII, and VIII. They are, temperature t, pressure p, specific volume v, condition, defined by quality x or superheat $s = (t - t_s)$, total heat h or H, and entropy n or N. A scheme in which it is possible to combine all these in one diagram is now to be described and illustrated, although the diagram in shape for regular use is not made a part of this book. In the Rankine cycle, adiabatic or constant-entropy expansion is the determining requirement, and especial emphasis is laid upon total heat. Entropy n and heat h are therefore made the coördinates of the diagram shown in Fig. 72, the Mollier diagram.

(*b*) THE MOLLIER DIAGRAM. — Fig. 72 is intended simply to illustrate this scheme; in a service diagram, a great many more of each

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kind of line would have to be drawn. With the entropy in constant-pressure heating from 32 deg. fahr. as base, and with the corresponding total heat h as ordinate, curves are drawn for particular, constant values of each of the other four quantities named above. The first line plotted is the saturation line AA, with N from col. 13, H from col. 8, of

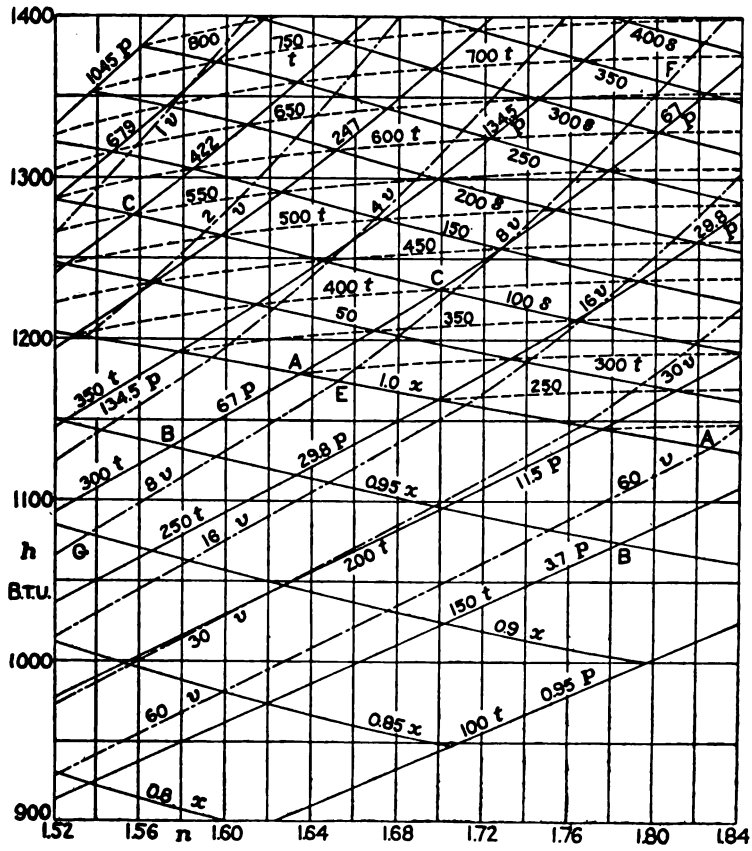

FIG. 72. — Outline of the Mollier Diagram.

Table II. Below this, or in the region of wet steam, the lines of constant pressure and of constant temperature are identical, and they are straight lines because total heat and entropy vary together in constant ratio during vaporization. On these lines like AB it is easy to locate points for particular values of x, and thus get the constant-quality lines BB. Here the p and t lines are spaced for simple-number values of t, rather than of p.

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For superheated steam, we have in Table VII intersections of lines of equal superheat, of equal temperature, and of equal pressure (the last corresponding to 10-deg. values of the saturation temperature), and can read off the total heat at each intersection. As an example, where pressure $p = 67.0$ ($t_s = 300$), superheat $s = 100$, temperature $t = 400$, we read $h = 1231.2$. The same intersection can readily be located in Table VIII, and the entropy read as $n = 1.6991$. These values of n and h are the coördinates of the point C, the intersection of the 67-lb. p line, the 100-deg. s line, and the 400-deg. t line in Fig. 72. With enough of these points, the three sets of curves can be drawn.

The lines of constant volume are laid out last. With a sufficient number of constant-pressure and constant-temperature lines drawn in the region of superheat, it is simplest to locate points from Table VI by interpolation between the isothermals in Fig. 72. For wet steam, we get x for a certain volume at a certain pressure, then find h for this x and go to the proper height on the p line in Fig. 72.

(*c*) THE SERVICE DIAGRAM. — In a regular reference diagram, the isothermals are commonly omitted, and the pressure lines are drawn for integral values of p, which gives them widely variant spacing, with numerous changes of interval. The volume curves differ so little in trend from the pressure curves, that if enough of both sets are put in for reasonably close interpolation, the mass of lines becomes rather confusing to the eye. Sometimes the diagram is printed in two colors, but a really accurate "register" of the two impressions seems to be hard to secure. To get both clearness and accuracy, the diagram must be to a large scale; and when there is room for this, it is exceedingly useful and convenient. Running down a line of constant entropy, we have right before us the changing pressure, volume, condition, and total heat, in adiabatic expansion.

(*d*) OTHER SCHEMES. — It is evident that any two of the six quantities named in Art. (*a*) might be used as coördinates, after the manner of Fig. 72, and curves be drawn for particular values of others. Thus in Table VII the coördinates are pressure and total heat, and lines are drawn for constant values of temperature and of superheat; on a larger diagram, constant-entropy lines could be added to advantage. In general, volume is the most difficult quantity to plot in a satisfactory way, especially in direct relation to pressure and temperature; with total heat for one coördinate, as in Fig. 72, it shows less extreme variation than in perhaps any other scheme of representation.

These special graphical methods are most useful when a great many determinations are to be made, as in some extended investigation.

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For the learner, the less concentrated methods exemplified in the preceding sections are of more importance; as illustrating and enforcing principle, they should be thoroughly understood before coming to short cuts and labor-saving devices; and a clear understanding of principle will remove all difficulties in the way of intelligent use of such special devices.

The Mollier diagram has its important field of usefulness in connection with the steam turbine. Examples of its application in analysis of performance and in design of fundamental proportions will be found in § 48 (*l*) and § 49 (*c*).

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CHAPTER V

ACTION OF THE STEAM IN THE ENGINE

§ 19. The Indicator Diagram

(*a*) Comparison of Actual with Ideal Diagram. — Continuing the line of the discussion in § 15, we now pass from the ideal steam diagram developed in Fig. 57 to the actual indicator diagram. The ideal diagram is modified in two principal directions: first, by resistance to the movement of the steam into and out of the cylinder; second, by the presence of clearance and compression. The thermal action of the cylinder walls has already been described in a general way in § 15 (*e*) and is more closely discussed and analyzed in §§ 22 to 25.

In Fig. 73, the indicator diagram ABCDEF represents the performance of the steam in the cylinder, and is to be compared with the circumscribed ideal diagram; which latter, however, is neither GHLQT nor JHLQP, but is shown in its true relative form in Figs. 74 and 75. The type of diagram in Fig. 73, chosen for illustrative purposes because the secondary effects now to be considered are all of good magnitude, is characteristic of the high-speed, noncondensing engine, with large clearance and with the steam distribution controlled by a single slide valve, so that high compression necessarily accompanies early cut-off. This is not an exact reproduction of an indicator diagram, because certain irregularities in outline, due to the instrument, are smoothed out; further, the curves of expansion and compression are taken to be simple equilateral hyperbolas.

(*b*) Reference Lines. — The indicator draws the outline ABCDEFA and the atmosphere line TQ. The lines drawn about the diagram by hand, to get Fig. 73, are as follows:

The vacuum line ON is parallel to the atmosphere line PQ, at the distance 14.7 lb. per sq. in., to scale, below it.

The steam-pressure line JK is located by measuring the reading of the pressure gage from PQ upward. If this gage is on the boiler, perhaps at a considerable distance from the engine, the pressure difference GA is likely to be considerably greater than in Fig. 73, often rising to 10 or 15 pounds.

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The end lines GM and KN touch the diagram, marking off the length MN which represents the stroke of the piston or, to a suitable scale, the volume displaced by the piston in one stroke, also called the nominal cylinder volume.

The distance MO is measured off to represent the clearance volume to this same scale, and the clearance line or pressure axis OJ is drawn. The clearance volume is made up of the space between the cylinder head and the piston (the crank being on dead center) together with the steam port or ports at one end. It is expressed as a fraction or percentage of the nominal volume, which is also the ratio of OM to MN.

Finally the expansion curve is produced upward to meet the line of boiler pressure at H and downward to the end line at L. Very commonly the $pv = C$ curve is drawn along the expansion curve, in coincidence at a point just after cut-off, in order to see how nearly the actual curve conforms to this simple "standard."

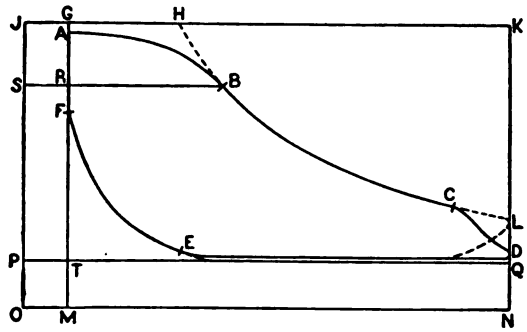

FIG. 73. — A Sample Indicator Diagram.

(c) ADMISSION. — This operation begins at F; but the filling of the clearance space is so closely allied with compression that it can better be considered in that connection, in Art. (i). Starting at A, with the clearance filled, we note that the initial pressure in the cylinder is less than the outside steam pressure, because of the various resistances to flow of steam. In magnitude, the drop GA here shown is such as might be got by having the pressure gage on the steam pipe, near the engine — a very proper arrangement when the performance of the engine alone is to be measured and represented. For the present, however, we shall not distinguish steam-pipe pressure from boiler pressure.

As the piston advances and the valve begins to close, the pressure falls off. Complete closure, or mechanical cut-off, is marked by the point B where the convex admission curve merges into the concave expansion curve — or where the curvature reverses in direction. But ex-

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pressed in terms of the amount of steam admitted, rather than by the position of the piston when the edge of the valve meets the edge of the port, the effective cut-off is at H: the volume JH measures, at boiler pressure, the total amount of steam present in the cylinder.

The admission line AB is really the resultant of the constant-pressure expansion GH in vaporization and the portion of the falling-pressure expansion from H to B. Because of all the resistances which the steam has to overcome in getting into the cylinder, there is a loss of available work equal to area GHBA. The action is of the nature of throttling, and at B the heat equivalent of this work has been added to the heat content of the steam.

(*d*) CUT-OFF. — There are two ways of locating this and other events in the steam distribution. From the point of view of mechanism, with emphasis upon the fact that the valve closes when the piston is at the distance RB from the beginning of the stroke, we say that the ratio of apparent cut-off by the valve is

$$C_{VA} = \frac{RB}{MN}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (110)$$

But in order to have a ratio of volumes, the clearance must be included; the volume back of the piston at cut-off is SB, while that at full stroke is ON. Then the real ratio of cut-off by the valve is

$$C_{VR} = \frac{SB}{ON}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (111)$$

All distances and volumes are expressed as fractions of the stroke or of the nominal cylinder volume, represented by MN, which is taken as unity. If we let i stand for the clearance fraction, or the ratio of OM to MN, the relation results,

$$C_{VR} = \frac{C_{VA} + i}{1 + i}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (112)$$

In the figure, $C_{VA} = 0.35$ and $i = 0.10$; then $C_{VR} = 0.45 \div 1.10 = 0.409$.

Besides the two ratios of cut-off by the valve, or of actual cut-off, there are two corresponding ratios for the effective or ideal cut-off at H — which is called "ideal" because, if there were no losses of pressure from the boiler, and if the valve could act instantaneously, this is where it would close in order to determine the admission of the same amount of steam into the cylinder as by the actual cut-off at B.

The apparent effective cut-off C_{EA} = GH/MN is very nearly what is called the "commercial cut-off" in the Code of Rules for Steam-Engine Tests of the American Society of Mechanical Engineers — see *Transactions*, Vol. 24, 1903, page 749: the definition is modified by requiring that H shall lie on the line of initial pressure, which touches

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the admission line AB at its highest portion. In other words, this commercial cut-off is the ratio of AK to MN in Fig. 74.

The effective real cut-off,

$$C_{ER} = \frac{JH}{ON}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (113)$$

is an important quantity, being the ratio of the initial volume of all the steam in the cylinder, if at boiler pressure, to its final volume when the stroke is completed.

(*e*) EXPANSION. — Of the shape of the expansion curve little need be said here, except to remark that it is seldom a true hyperbola, either rising above or falling below that curve, although not by any large amount under average conditions. In addition to the general description of the thermal action of the cylinder walls given in § 15 (*e*), and in anticipation of the fuller discussion in the latter part of the chapter, it is now appropriate to describe briefly the means employed to control and minimize this action.

By compounding the engine, or by dividing the expansion from boiler pressure to exhaust pressure into two or more successive stages in separate cylinders, the range of pressure and of expansion in the individual cylinder is diminished, with a marked decrease in initial condensation.

By the use of a steam jacket, the cylinder walls are kept hotter and drier, and thus the freedom of heat transfer between the entering steam and the metal is decreased. The jacket consists of an annular space around the barrel or body of the cylinder, and of suitable hollows in the heads — see Fig. 203. It is filled with steam from the boiler, and provision must be made for draining out the water of condensation. In the large, low-pressure cylinders of multiple-expansion engines, the pressure of the jacket steam is commonly reduced, chiefly from considerations of strength.

The use of superheated steam has for its main purpose the diminution of cylinder losses, which is effected by decreasing the freedom of heat transfer, through keeping the steam drier. Only with a very large excess of temperature does the steam remain superheated at cut-off and during expansion, so that there is little chance to utilize the greater available temperature range of the heat added in superheating, which is shown in Fig. 56.

Increase of size, of speed, and of the quantity of steam admitted (later cut-off) all tend to diminish the amount or proportion of steam initially condensed. As regards the effect of these various devices and conditions upon the shape of the expansion curve, the general principle prevails that whatever diminishes the heat returnable during expansion

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tends to make the curve drop more rapidly after cut-off, and vice versa.

It may be well to call attention to the point of usage, that in the language of the steam engine the term "expansion" is limited in application to the expansion with fall of pressure, from H or B to C in Fig. 73. Of course, the constant-pressure expansion in vaporization, active at the surface of the water in the boiler and transmitted forward until it becomes effective upon the piston, is equally entitled to the name; but it is usual to let "admission" cover this action.

(*f*) RELEASE AND EXHAUST. — The port is opened for exhaust before the stroke is completed, and the pressure falls along the release line CD as the steam is discharged. In the case of our sample diagram, about half the steam escapes from C to D, the rest during the constant-pressure exhaust from D to E. On account of the early release at C, there is a loss of work represented by the area CLD. This loss can hardly be avoided, for if the valve was kept closed till the end of the stroke, it could not be opened quickly enough to obviate a throttling of the release such as is shown by the dotted curve from L.

The same causes that produce a loss of pressure during admission are responsible for the excess back pressure, which lifts the exhaust-line DE above PQ, and deducts another small portion from the area of the ideal diagram.

(*g*) CLEARANCE AND COMPRESSION. — After exhaust ends, at E, the low-pressure steam retained in the cylinder is compressed up to F by the piston; then fresh steam is let in, and the pressure in the clearance space is raised up to A before the piston makes any appreciable forward movement. The operation of admission consists, therefore, of two parts, the first the filling of the clearance space while the piston is practically standing still; the second the filling of the cylinder back of the piston as the latter advances, out to cut-off. Taking JHLQP as the ideal diagram (exactly like Fig. 57), the area JGFEP appears to be subtracted from it as the result of compression: but since the compressed steam so nearly fills the clearance space,* the volume of new steam is much less than JH (but little more than GH), so that the efficiency, measured by the ratio of work done to steam used, is not so very different from what it would be for the full diagram, back to the axis OJ.

(*h*) WORKING STEAM AND CLEARANCE STEAM. — A clear understanding of the effects of clearance and compression can be gained by

* For the sake of simplicity, the effect of cylinder-wall action upon the volume of the clearance steam, during expansion, is not taken into account here: note remark in Art. (*k*).

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thinking of the steam in the cylinder as made up of two parts — the live or working steam which enters the cylinder to do work and escapes after the work is done, and the dead or clearance steam which, as a certain quantity, remains in the cylinder indefinitely, alternately expanding and contracting as the pressure changes. We can even imagine the latter body of steam to be separated from the working steam by a sort of light diaphragm, which will move back and forth in the cylinder like a loose auxiliary piston. When compression begins, at E in Fig. 74, this diaphragm is against the piston, and so remains up to F; then the piston stops, and the compression of the dead steam is continued along the curve FU by the fresh entering steam.

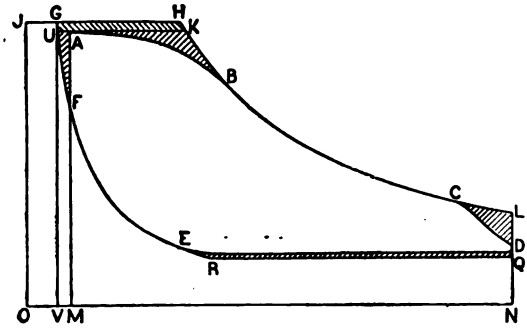

FIG. 74. — Kinetic Losses of Work Effect.

(*i*) FILLING THE CLEARANCE SPACE. — The raising of the pressure from F to A, Fig. 73 or Fig. 74, takes place while the valve is open but a very little way. The live steam, flowing through this narrow opening into a space where the pressure is lower, undergoes a regular throttling action, with some consequent loss of available energy. When the supplementary compression up to initial pressure is completed, the volume OV, Fig. 74, is filled with old steam and the volume VM with new steam. The latter, if admitted to drive the piston forward, could have done the work represented by the rectangle UAMV. Actually, it has done upon the clearance steam the pressure work FUVM (which can come back upon the piston during expansion), and has wasted the area AFU in noneffective acceleration of itself.

(*j*) KINETIC LOSSES. — It appears now that the area AFU is to be added to the losses due to the moving of the steam, described in Arts. (*c*) and (*f*). The areas representing all these kinetic losses are shaded in Fig. 74. The steam-pipe effect GHKU shows pressure that was used up, in small part to produce the initial velocity of flow along the pipe, in larger part to maintain this flow against surface friction and the retarding influence of bends and other obstructions. Area AKB is

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the loss caused by the throttling effect of the slide valve. The triangular figure CLD is only a small portion added to the loss of available energy due to incomplete expansion — all of this unused work (see area CHD, Fig. 57) going to produce a high velocity of the exhaust steam. Because of the resistance of port and exhaust pipe, area DERQ is added to the losses incurred in moving the steam.

(*k*) ACTION IN EXPANSION. — As the pressure falls, toward and after cut-off, the clearance steam expands with the working steam. We now assume, as a rough first approximation, that the curve UFE is retraced in expansion, down to the lowest pressure. Actually, because of condensation by the cylinder walls and shrinkage in volume, of which the clearance steam suffers its full share, the latter expands along a curve inside of UFE: further discussion of this matter will be found in § 23, where the behavior of the clearance steam is more fully considered. With this steam retracing UFE, the effective vol-

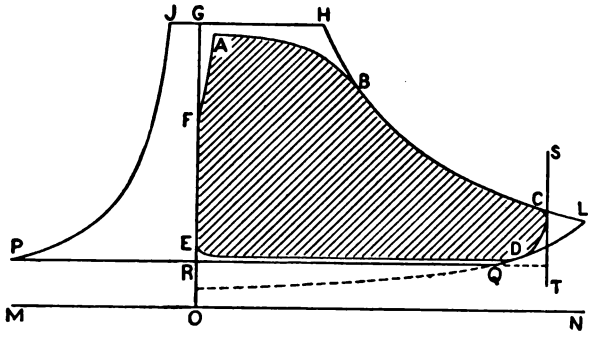

FIG. 75. — The Diagram of True Expansion.

ume of the active steam must be measured, at any pressure, from this curve — or from the position of an imaginary partition between the two bodies of steam. Since the reversed compression curve extends so far to the right, especially after it gets below the level of the release point C, it diminishes very considerably the amount of useful expansion realized in the cylinder.

This action is most clearly shown by Fig. 75, which is derived from Fig. 74 through the rectification of the reference curve GFE, with horizontal shifting of the volume abscissas. Bringing the hyperbola GFE (Fig. 74) to the straight line GFE (Fig. 75), throws the zero line JO (Fig. 74) out to the hyperbola JP (Fig. 75). The scheme is most concisely defined, perhaps, by saying that in Fig. 75 volumes are shown as if measured, in both directions, from the imaginary dividing plane between the two bodies of steam, not from a fixed cylinder end. The

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performance of the clearance steam is now shown by the curve JP, and ceases to be of much interest.

(*l*) Effective Expansion. — The realized performance of the working steam, freed from complication, is now given by the distorted diagram ABCDEF, shaded in Fig. 75 and in shape for direct comparison with the ideal diagram of Fig. 57. The true ratio of expansion is got by comparing the maximum volume RT with the initial volume GH. In Fig. 73, the apparent effective cut-off ratio, GH to MN, is 0.25; the ratio of volumes, including clearance, JH to ON, is 0.318; but the true effective ratio, GH to RT in Fig. 75, is 0.354, or the realized ratio of expansion is only 2.82.

This last ratio is the most important piece of definite information supplied by the transformed diagram in Fig. 75. After we have gained from the latter the clearer idea of the performance of the working steam which it gives, we can go back to the original indicator diagram, and readily get from it this same ratio of effective expansion. By holding a scale parallel to the atmosphere line in Fig. 74 and moving it up and down — as with the diagram fastened down on the drawing board and the scale against the T square — it is easy to find by trial the greatest horizontal distance between the compression curve EF and the release line CD; and this maximum distance is the same thing as RT in Fig. 75. The initial volume GH is directly measurable on Fig. 74.

(*m*) Form of the Compression Curve. — With a moderate amount of compression, say not more than enough to raise F halfway from T to G on Fig. 73, this curve agrees fairly with the hyperbola. But when compression begins early in the return stroke, so that the relatively small weight of clearance steam is raised well above the temperature of the cylinder walls quite an appreciable time before fresh steam enters, there will be a marked abstraction of heat from this steam, with shrinkage in the value of the pv product and lowering of the upper part of the curve and of the point F. In extreme cases, the steam at F may be wetter than the whole body of steam at cut-off. This matter is more fully set forth in § 23.

(*n*) Information from the Diagram. — From the indicator diagram itself — that is, without the further knowledge obtained from a measurement of steam used — the following information can be derived:

The form of the diagram shows the working of the valve gear, a matter which is not considered here, but is taken up in Chapter VIII. Also, from the shape of the curves of expansion and compression it is possible to draw some roughly approximate inferences as to thermal interactions, and perhaps as to leakage of steam.

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Having found from the diagram the mean effective pressure exerted upon the piston, it is easy to calculate the work done by the steam, or the indicated horse-power of the engine. The actually delivered or effective power — often called the brake horse-power, because it can perhaps be most exactly measured by means of the friction brake — is less than the indicated power by the rate of work absorption within the machine, against its own friction. The ratio of effective to indicated power is called the mechanical efficiency of the engine.

From the fact that the properties of saturated steam, including specific volume, are fixed when the pressure is known, the engine can be made to serve as a volume-meter, to determine the quantity (weight) of steam used. Unfortunately, this measures only what is present in the cylinder as vapor, not what is also present as water, because of initial condensation. Between indicated and actual steam consumption there is always a considerable discrepancy. A true and reliable determination can be made only by a test in which the actual, total steam used is measured, either as water going to the boiler or, preferably, as water formed by condensing the exhaust in a surface condenser.

The subject of indicated power and steam consumption will be taken up in § 21, while the closer study of the thermal action of the cylinder walls is carried forward in § 22. As a preparation for these discussions, a simple presentation of fact and principle in regard to the compound engine will now be made.

§ 20. The Compound Engine

(*a*) OBJECTS OF COMPOUNDING. — These can best be made clear with the help of an example. In Fig. 76, let the outline ABDEFG be the ideal steam diagram (like Fig. 57) for an engine to work between the pressure limits $p_1 = 120$ lb., $p_0 = 1.6$ lb. In its length and narrowness, with a high ratio of maximum to average pressure, this diagram is open to the objections urged against the Carnot cycle with air in § 8 (*i*), although it has the redeeming feature of a low back pressure. From the mechanical viewpoint, if the whole operation is carried out in a cylinder big enough to contain the final steam volume GF, the engine will have to be tremendously strong and heavy in order to carry the high pressure at the beginning of the stroke. From the thermal side, with the range of temperature from t_1 to t_0, and with the very short cut-off, ratio $e = AB/GF$, the initial condensation by the cylinder walls will be excessive.

These difficulties are very effectively overcome by dividing the expansion into steps or stages. The line JD is drawn at such a height,

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in Fig. 76, that it divides the area ABDEFG into about equal parts — in the final adjustment, areas ABCHJ and JDEFG are made equal. The work above this line is performed in the small, high-pressure cylinder; that below, in the large, low-pressure cylinder. Between the cylinders is placed an intermediate vessel or reservoir called the receiver, now assumed to be so large that the pressure within it shows no cyclical fluctuation, due to the inflow and outflow of steam, but remains constant at the value represented by the height of line JD. The same considerations that demand and justify incomplete expansion for the whole engine — see § 15 (*g*) — are influential, although in lower degree, for the high-pressure cylinder. Instead of giving it the volume JD, we draw a line at PH to fix its volume; the exact location of this line is a

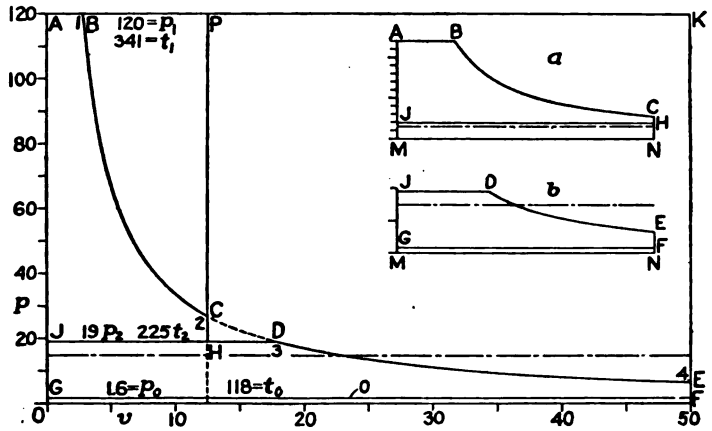

FIG. 76. — Ideal Diagram for the Compound Engine.

matter of judgment, involving as it does the sacrifice of the work-area CDH, through the pressure drop CH, which is called "receiver drop."

The objects attained by this scheme for the effective utilization of a high ratio of expansion are, then, to get more uniform and less widely variant forces in the machine, and to keep the harmful action of the cylinder walls within reasonable bounds.

(*b*) CYLINDER RATIO AND SEPARATE DIAGRAMS. — Almost universally, different sizes or volumes of cylinder are obtained by the use of different diameters with the same stroke-length; in a very few special or freakish designs, the strokes have also been different. In Fig. 76, JH is made one-fourth of GF, or the cylinder ratio GF to JH is fixed at 4 to 1; then the diameters are as 2 to 1. At *a* and *b*, the two parts of the main figure are changed to a form corresponding with ordinary indicator diagrams. An arbitrary base length MN is chosen, and the

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low-pressure diagram JDEFG is shortened to this length, at *b*, but with no change of vertical measurement. In the stretching of the high-pressure diagram ABCHJ to the same length as the other, at *a*, there is an effect equivalent to the multiplication of its abscissas by the cylinder ratio, here 4. To compensate for this, and have the two diagrams *a* and *b* represent work to the same scale, the original ordinates of ABCHJ must be divided by the same ratio 4.

Now there are two ways of looking at this high-pressure diagram with shortened ordinates. If the original scale is, say, 20 lb. to the inch, the new diagram will have a scale of 80 lb. to the inch when representing pressures on the high-pressure piston. Or, we may measure the shortened ordinates to the 20-lb. scale, and say that they then represent equivalent pressures on the low-pressure piston. In numerical expression, with the particular proportions, a specific pressure of 4 lb. (per sq. in.) on the small piston will produce the same total force and do the same work as 1 lb. per sq. in. on the large piston. According to the latter concept, with transfer of pressure and work from one piston to another, diagram *a* is said to be reduced to the low-pressure piston.

Actual indicator diagrams follow, of course, the first idea above, in that each represents directly the variable pressure upon its own piston. It is neither convenient nor, in many cases, possible to choose springs whose scales are in the exact ratio of the cylinder volumes: but the idea of "reduction" of pressure from one piston to another has some important applications, as will appear later.

(*c*) ARRANGEMENT AND WORKING OF ENGINES. — In the stricter technical usage, the adjective "compound" is limited to engines with two stages of expansion, as in Fig. 76; "multiple-expansion" is the more general term. An engine with three stages is called a triple expansion, one with four stages a quadruple-expansion. The highest number of stages ever used, in a few exceptional marine engines, is five. The degree of compounding (going back to the more general use of this term), is chiefly determined by the boiler pressure. In marine practice, with engines always run condensing, the ranges are about as follows, the line of historical progress being from the lower to the higher value in each case:

Simple engines	30 to 70 lb. by gage
Compound	80 to 120 " " "
Triple	140 to 180 " " "
Quadruple	200 to 250 " " "

In stationary practice, there is a tendency toward higher pressures for the same classes of engines, condensing compounds being run with

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boiler-gage pressures of 120 to 150 lb.; while the compound locomotive, necessarily noncondensing, uses steam at from 200 to 225 lb.

In the matter of getting the steam from one cylinder to the other, there are two typical cases. First, if the two pistons begin and end their strokes together, so that steam can pass from the higher directly into the lower cylinder, it is possible to get along without a receiver: this is called the direct-expansion compound. Second, if the two strokes are not timed together, as when there are two cranks at right angles to each other, the receiver is essential, and we call this a receiver-compound engine.

Engines of the first class may have the cylinders placed in line (tandem) or side by side (parallel); and the connecting passages and low-pressure steam chest may amount to a considerable receiver space. An engine having cylinders side by side and cranks at an angle with each other is called a cross-compound.

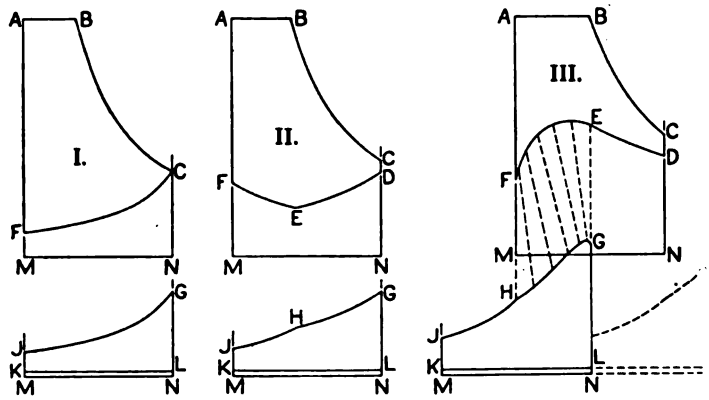

FIG. 77. — Typical Cases of Receiver Action; Ideal Diagrams, for cylinders having no clearance space.

(*d*) LINES OF INTERMEDIATE PRESSURE. — The characteristic effects of the changes in arrangement just described are illustrated in Fig. 77: all the variations occur in the lines which correspond to CH–HJ and JD in Fig. 76, that is, in the lines of intermediate pressure, between the cylinders and involving the receiver. The low diagram is placed beneath the high, so that coincident timing appears directly in the first two cases and can readily be shown in the third.

Case I. Direct-expansion engine, with no receiver. At C and G begins a common expansion in the two cylinders, which must continue clear to the end of the stroke, to F and J, in the ideal case of no clearance. In an actual engine there is low-pressure cut-off near the end of the stroke, and a short compression into the high-pressure clearance.

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Case II. Direct-expansion engine with a receiver. The receiver drop CD is followed by a common expansion DE and GH, to low-pressure cut-off at E and H. Then there is compression in the high-pressure cylinder and the receiver along EF and expansion in the low-pressure cylinder along HJ.

Case III. Receiver-compound engine with cranks at right angles. Receiver drop CD is followed by compression in the high-pressure cylinder and the receiver, along DE, until the low-pressure cylinder is ready for admission at E and G: then comes a common expansion EF and GH, to low-pressure cut-off, which, as the simplest case, is taken to occur just at half-stroke.

In cases II and III, the pressure at D is the result of a mixing of two bodies of steam, that in the small cylinder at pressure C and that in the receiver at pressure F. The most striking thing shown in Fig. 77 is the characteristic difference between the forms of the high-pressure exhaust line DEF in cases II and III, which clearly distinguishes the two types of engine.

For the detailed development of these diagrams, in the way of quantitative relation and the exact form of the curves, the reader is referred to STEAM ENGINE, Vol. II, Chapter XI, pages 449 to 476. In actual engines the ideal curves of intermediate pressure are very considerably modified by resistances to flow and similar influences, always in the direction of a more uniform pressure in the receiver.

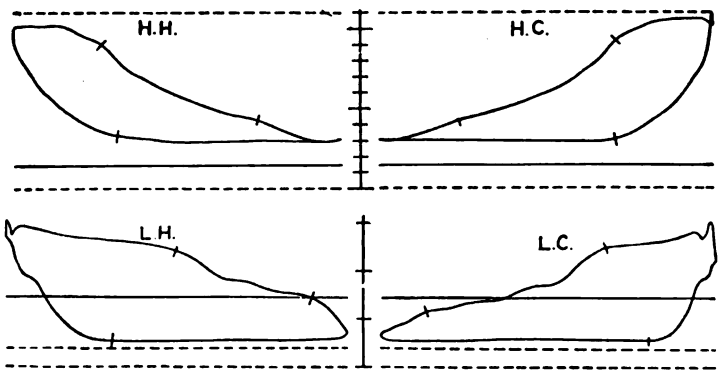

FIG. 78. — Indicator Diagrams from a Compound Engine. Steam pressure near engine, 96 lb. by gage or 111 lb. absolute; condenser pressure, 22.2 in. mercury of vacuum or 3.9 lb. absolute; both pressure lines drawn on diagrams. Speed, 250 r.p.m. Indicator springs, 60 lb. scale in high-pressure, 20 lb. in low-pressure cylinder.

(*e*) INDICATOR DIAGRAMS. — A sample set of diagrams from a compound engine, which is of the same type as the simple engine rep-

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resented by Fig. 73, is given in Fig. 78. The arrangement of the cylinders is outlined in Fig. 79, which also carries the essential dimensions. The size of a multiple-expansion engine is defined by giving the diameters in sequence, then the stroke; this is a 13 and 20 by 16 inch tandem compound. The arrows on Fig. 79 show the sequence of the cylinder ends in steam distribution; there is, of course, only a single passage from the high-pressure exhaust port to the low-pressure steam chest, but the steam from either end of the small cylinder goes into the opposite end of the large cylinder. The condenser pressure is found by subtracting the 22.2 in. of gage reading from 29.9 inches, the standard atmosphere, and reducing 7.7 inches to pounds of absolute pressure; the vacuum is very poor in this case, for it ought to be at least 26 or 27 inches with proper working of the condenser. In the notation used to designate the diagrams, the first letter is for the cylinder, high or low, the second for the end, head or crank. The steam-distribution events are marked on Fig. 78, and a considerable inequality in the low-pressure cut-offs is apparent.

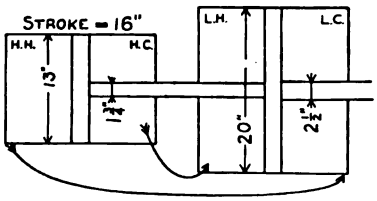

FIG. 79. — Cylinder Arrangement of Tandem Engine. Clearance ratios, high head, 0.16, high crank, 0.12, low head, 0.08, low crank, 0.06.

(f) COMBINED DIAGRAMS. — A most useful means of showing the action of the steam in a compound engine is the combined diagram, in which the separate indicator diagrams are brought to the same scales of pressure and of volume, and referred to the same axes — the operation being the reverse of the derivation of separate diagrams in Fig. 76. Two methods of combining will now be illustrated, the first fully defined by the statement just made, the second embodying in addition the idea of Fig. 75.

In part I of Fig. 80, the indicator diagrams are prepared by dividing the length of each into ten equal parts, and erecting ordinates at the division points. At II the volumes, first of the clearances, next of the cylinders (nominal), are laid out from OP to a convenient scale, and the volumes M_1N_1, M_2N_2, are divided by lines corresponding to those on the diagrams in I. In this case, the pressure scale is the same as that of the low-pressure diagram, so that ordinates from the latter can be transferred directly; while those from the high-pressure diagram must

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be multiplied by three, or measured off three times. Using, if necessary, extra ordinates at the ends where the pressures are changing rapidly, we get a series of points through which the new curves can be traced, giving the result shown by the full-line figures.

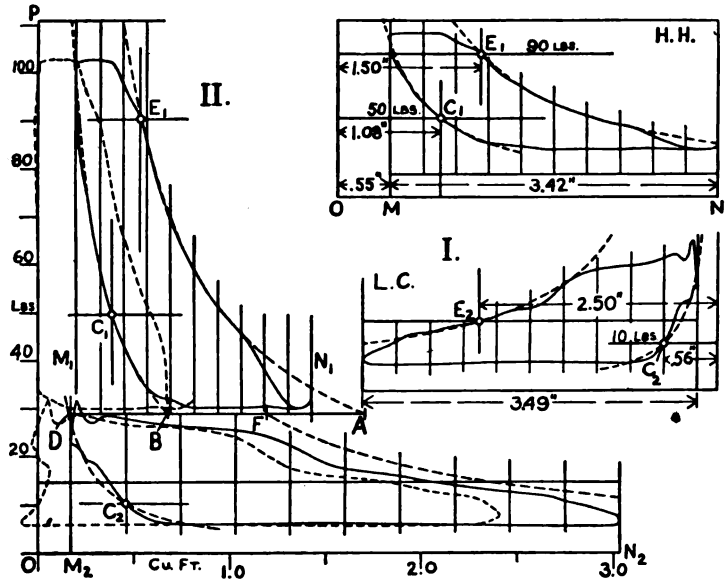

FIG. 80. — Diagrams Combined on Clearance Lines.

On the original diagrams, hyperbolas are drawn through chosen points, marked E and C, on the expansion and compression curves: on the combined diagram, three of these are reproduced, all except that through E_2. Instead of the latter, the curve through E_1 is continued, in effect, so as to get a correct *pv* measure of the continuity of the expansion. A horizontal line AD is drawn between the diagrams, the two upper hyperbolas cutting it at A and B, while that through C_2 meets it at D; then the length DF is made equal to BA, and the expansion hyperbola is continued from F. Obviously, this is nearly as effective as the method of Fig. 75; for while it does not get rid of the clearance steam, it does eliminate the difference between the two quantities of this steam in the successive cylinders.

Diagrams on the rectified compression hyperbolas, similar to Fig. 75, are dotted in on Fig. 80, but without any of the construction used in getting them being given. This would consist of a lot of horizontal lines along which the volumes, measured from the reference curves C_1B, DC_2, would be laid off from OP. The effect of indicator inertia

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upon the low-pressure compression curves is strikingly shown by the peculiar heel on the derived diagram: the indicator piston at first lags behind the rising pressure, then swings ahead of it, and keeps oscillating about the true pressure until its energy is absorbed in friction work. The hyperbola through C_2 should follow the mean of these waves.

(*g*) Direct Combination on the Compression Lines. — This is illustrated in Fig. 81, where a graphical construction is used for trans-

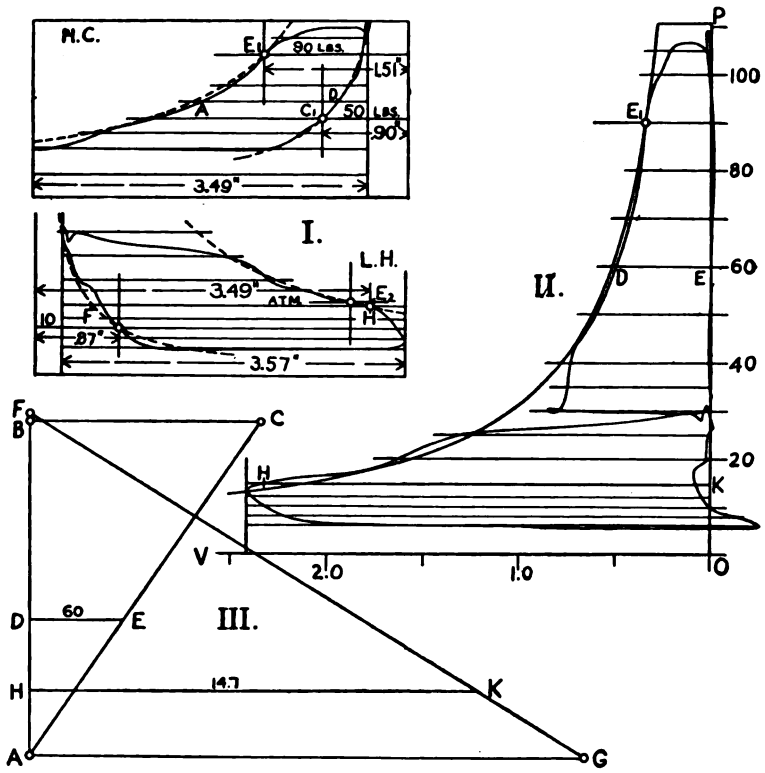

Fig. 81. — Diagrams Combined on Compression Curves.

forming the volume ordinates to their new scales. The first step is to draw a number of similarly spaced abscissa lines on the indicator diagrams and on the plane of the combined figure, at II. In the proportion diagram at III, AB is the actual length of the high-pressure diagram and BC is the corresponding volume to the scale of II, the same thing as M_1N_1 in Fig. 80. After the diagonal AC is drawn, any abscissa as AD in I is laid off from A to D in III, and the intercept DE is the converted length, ready to be transferred to the position DE in II.

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Similarly, FA is the length of the low-pressure diagram, AG is the corresponding volume to the scale of II, and F is the origin: FH is measured from F and HK is ready to be laid off in II. It is more convenient to use these proportion diagrams than to work through the first method of combination, as in Fig. 80.

In Fig. 81, part II, a hyperbola is drawn through E_1, and cuts under the low-pressure expansion curve; but in Fig. 80 the hyperbola from F is well above its expansion curve. It appears then that the steam is not divided between the two ends of the low-pressure cylinder in the same ratio as between the two ends of the high-pressure cylinder. The only way to get a fair criterion of the total expansion is to draw a mean combined diagram.

(*h*) THE AVERAGE COMBINED DIAGRAM. — In Fig. 82, the diagrams referred to the rectified compression hyperbolas are reproduced from Figs. 80 and 81 (in dotted line), with the irregularities on the compression side left out, and mean curves are partly drawn (in full line). Besides this averaging of the two ends, there is a change of volume scale, such that the diagram now represents the performance of one pound of steam: the method of determining the limiting volumes for this unit diagram is given later, in § 21 (*l*), since it involves the rate of steam consumption by the engine.

On Fig. 82 several reference curves are drawn, and at the right is a supplementary diagram of steam quality or condition during the respective operations of expansion. The hyperbola ABC is the first curve laid out, being made coincident with the high-pressure expansion curve just after cut-off. The low-pressure expansion curve rises above the hyperbola, which indicates considerable leakage through the high-pressure cylinder: it is normal for an unjacketed compound to show a shrinkage in the product pv from the high to the low cylinder, perhaps as much as 10 or 15 per cent. The curve DE is the line of constant steam weight for one pound of steam, or it is the saturation volume s plotted right from Table II. This is really the most useful simple curve that can be drawn for a reference line. An adiabatic from D would also be of interest, as the expansion line of the ideal Rankine cycle; but drawing it generally leads to an unpleasing confusion of lines along the low-pressure diagram.

The quality curves FF and GG are got by taking x as the ratio of HK to HL, and laying it off to the scale at the bottom of the figure. Of course, this is only an approximation, less correct as the curves conform less closely to $pv = C$ and as the expansion and compression curves differ in form. It is better to draw the x curve for the total steam in the cylinder, including the clearance steam, as is done in Fig.

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115, for instance. An example of such curves for the common form of combined diagram, like Fig. 80, but on a stroke instead of a pressure base, will be found in Fig. 136.

Finally, the hyperbola is plotted on the x diagram in curve A′C′

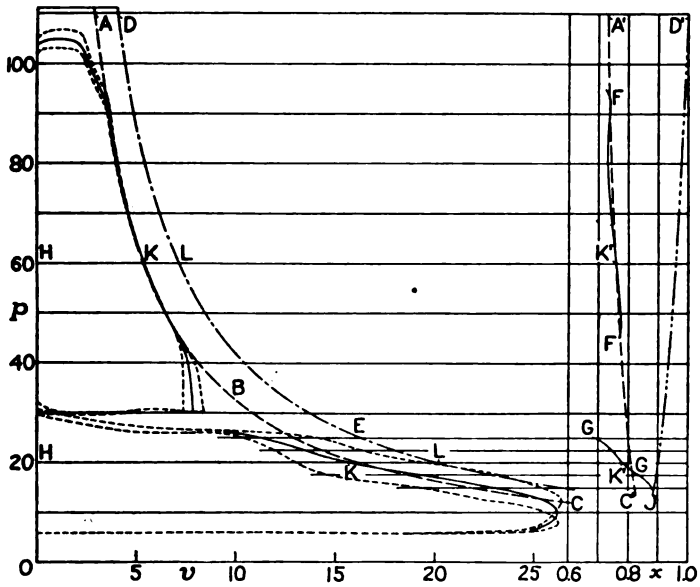

FIG. 82. — Mean Diagram for One Pound of Steam, from Figs. 80 and 81.

(compare Fig. 52), and the adiabatic as D′J; here the curves are much clearer and less confused than on the pressure-volume diagram at its low-pressure end.

§ 21. Horse Power and Steam Consumption

(*a*) MEAN EFFECTIVE PRESSURE. — During the forward stroke, from M to N in Fig. 83, the steam does upon the piston the amount of work represented by the area ABCDNM; reducing this figure to the equivalent rectangle GHNM gives the mean total or forward pressure p_{mf} = GM. Similarly, during the return stroke the work done by the piston upon the steam is DEFMN, and the corresponding mean-back pressure is p_{mb} or MK. Then the effective work, area ABCDEF or area GHLK, would be done by the action of the mean effective pressure p_m or GK upon the piston through one stroke.

In practice, p_m is not got by subtracting p_{mb} from p_{mf}, but is found by measuring the area of the enclosed diagram ABCDEF with a

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planimeter. Dividing the area by the length gives the mean height, and multiplying this by the pressure scale of the ordinate gives the m.e.p. If a planimeter is not available, it is necessary to divide the diagram into a number of narrow vertical strips (of equal width), and take the average of the middle ordinates of the strips, measuring them as intercepts between the curves ABCD and FED.

(*b*) WORK PER REVOLUTION. — Although p_m actually represents the difference between the two quantities of work done upon the piston in the two strokes which make up one revolution (one work is positive, the other negative), it is treated as if it were simply an unbalanced pressure acting upon the piston through the forward stroke. Then if A is the area of the piston in square inches and S the length of the stroke in inches, the work done in one end of the cylinder per revolution is,

$$U = p_m A \frac{S}{12} \text{ ft. lb.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (114)$$

In the other end of the cylinder there is done a similar amount of effective work, but generally not quite the same: how nearly alike the two m.e.p.'s will be depends upon valve action, and they may be very different if the valve gear is in bad adjustment; further, the area of the piston is reduced on one side by the cross section of the piston rod, so that generally the two A's are not the same. In any case, the sum of the two separate U's gives the total work per revolution.

Note that the use of the ordinary m.e.p. does not give the work per stroke: to get this, we should have to subtract from the forward-pressure work on one face of the piston the simultaneous back-pressure work on the other face. But while the separate works per stroke would not, in general, be the same as the works in the two cylinder ends, the sum of either two would give the same total work.

(*c*) INDICATED HORSE-POWER. — Letting N be the number of revolutions or of double strokes per minute, the indicated horse-power developed in one end of the cylinder is

$$H = \frac{p_m A S N}{12 \times 33000} . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (115)$$

Several partial i.h.p.'s having been found, as for the head end and crank end of the cylinder, the total power is got by taking their sum.

The constant part of the horse-power formula,

$$C = \frac{AS}{12 \times 33000}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (116)$$

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is called the engine constant or cylinder constant. It can be worked out once for all in any particular case, and made a matter of record for the engine.

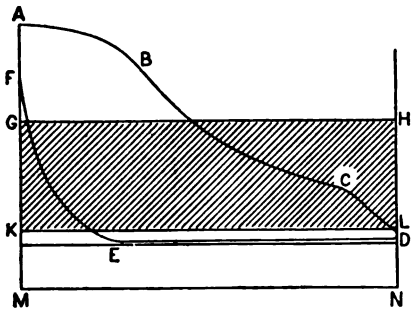

FIG. 83. — Gross and Effective Work.

(*d*) CYLINDER CONSTANTS. — In Table 9, the first column is diameter D in inches, the second is the area A of this circle in square inches. In column 3, V_0 is the volume in cubic feet of a cylinder D inches in diameter and ten inches long; and C_0, column 4, is the engine constant, Eq. (116), for each ten inches of stroke length. To get the volume or displacement V in cubic feet, for any cylinder of diameter D and stroke length S, take V_0 from column 3 and multiply it by $S/10$: to find the engine constant, follow the same procedure with C_0.

The small diameters in the first part of Table 9 are for the piston rod; their values are to be subtracted from those for the full piston diameter, before multiplying by $S/10$.

EXAMPLE 24. — Find piston areas, displacements, and cylinder constants for an engine 14 in. diameter by 15 in. stroke, with $2\frac{1}{4}$ in. piston rod — or of a $14 \times 15 - 2\frac{1}{4}$ in. engine.

The areas, from column 2 of Table 9, are

$$\text{Head end, } A_h = 153.94 \text{ sq. in.;}$$
$$\text{Crank end, } A_c = 153.94 - 4.00 = 149.94 \text{ sq. in.}$$

The volumes, taking V_0 from column 3 and with $S/10 = 1.5$, are

$$\text{Head end, } V_h = 0.8908 \times 1.5 = 1.336 \text{ cu. ft.;}$$
$$\text{Crank end, } V_c = (0.8908 - 0.0230) \times 1.5$$
$$= 0.8678 \times 1.5 = 1.301 \text{ cu. ft.}$$

Following the same method with C_0, the engine constants are found to be,

$$\text{Head end, } C_h = 0.003887 \times 1.5 = 0.005831;$$
$$\text{Crank end, } C_c = (0.003887 - 0.000100) \times 1.5$$
$$= 0.003787 \times 1.5 = 0.005681.$$

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TABLE 9. CYLINDER CONSTANTS.

D	A	V_0	C_0	D	A	V_0	C_0
1	.79	.0045	.000020	14	153.94	.8908	.003887
1⅛	.99	58	25	14½	165.13	.9556	4170
1¼	1.23	71	31	15	176.72	1.0227	4463
1⅜	1.49	86	38	15½	188.69	1.0920	4762
1½	1.77	.0102	.000045	16	201.06	1.1636	.005077
1⅝	2.07	120	52	17	226.98	1.3135	5732
1¾	2.41	139	61	18	254.47	1.4726	6426
1⅞	2.76	160	70	19	283.53	1.6408	7160
				20	314.16	1.8181	7933
2	3.14	.0182	.000079				
2⅛	3.55	205	90	21	364.4	2.004	.00875
2¼	4.00	230	.000100	22	380.1	2.200	960
2⅜	4.43	256	112	23	415.5	2.404	.01049
2½	4.91	.0284	.000124	24	452.4	2.618	1142
2⅝	5.41	313	137	25	490.9	2.841	1240
2¾	5.94	344	150	26	530.9	3.073	.01341
2⅞	6.49	376	164	27	572.6	3.313	1446
				28	615.8	3.563	1555
3	7.07	.0409	.000179	29	660.5	3.823	1668
3⅛	7.67	444	194	30	706.9	4.091	1785
3¼	8.30	480	210				
3⅜	8.95	518	226	31	754.8	4.368	.01906
3½	9.21	.0557	.000243	32	804.2	4.654	2031
3⅝	10.32	600	261	33	855.3	4.950	2160
3¾	11.05	639	279	34	907.9	5.254	2293
3⅞	11.79	683	298	35	962.1	5.568	2430
				36	1017.9	5.891	.02570
4	12.57	.0727	.000317	37	1075.2	6.222	2715
4½	14.19	.0821	358	38	1134.1	6.563	2864
4½	15.90	.0920	402	39	1194.6	6.913	3017
4¾	17.72	.1026	448	40	1256.6	7.272	3173
5	19.64	.1136	.000496				
5½	21.65	.1253	547	41	1320.3	7.640	.03334
5½	23.76	.1375	600	42	1385.4	8.018	3499
5¾	26.00	.1503	656	43	1452.2	8.404	3667
				44	1520.5	8.799	3840
6	28.27	.1636	.000714	45	1590.4	9.204	4016
6½	33.18	.1920	838	46	1661.9	9.618	.04197
7	38.49	.2227	972	47	1734.9	10.040	4381
7½	44.18	.2557	.001116	48	1809.6	10.472	4570
8	50.27	.2909	1269	49	1885.7	10.913	4762
8½	56.75	.3284	1433	50	1963.5	11.363	4958
9	63.64	.3683	1606				
9½	70.88	.4102	1790	51	2042.8	11.822	.05159
				52	2123.7	12.290	5363
10	78.54	.4545	.001983	53	2206.2	12.767	5571
10½	86.59	.5011	2187	54	2290.2	13.254	5783
11	95.03	.5500	2400	55	2375.8	13.749	6000
11½	103.87	.6011	2623	56	2463.0	14.254	.06220
12	113.10	.6545	.002856	57	2551.8	14.767	.06444
12½	122.72	.7102	3099	58	2642.1	15.290	6672
13	132.73	.7681	3352	59	2734.0	15.822	6904
13½	143.14	.8284	3615	60	2827.4	16.363	7140

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TABLE 9. — *Continued.*

D	*A*	V_0	C_0	*D*	*A*	V_0	C_0
61	2922.5	16.912	.07380	81	5153.0	29.821	.13663
62	3019.1	17.472	7624	82	5281.0	30.561	.13336
63	3117.2	18.040	7872	83	5410.6	31.311	.13013
64	3217.0	18.617	8124	84	5541.8	32.070	.13994
65	3318.3	19.203	8380	85	5674.5	32.839	.14330
66	3421.2	19.799	.08639	86	5808.8	33.616	.14669
67	3525.7	20.403	8903	87	5944.7	34.402	.15012
68	3631.7	21.017	9171	88	6082.1	35.198	.15359
69	3739.3	21.639	9443	89	6221.1	36.002	.15710
70	3848.5	22.271	9718	90	6361.7	36.816	.16065
71	3959.2	22.912	.09998	91	6503.9	37.638	.16424
72	4071.5	23.562	.10282	92	6647.6	38.470	.16787
73	4185.4	24.221	.10569	93	6792.9	39.311	.17154
74	4300.8	24.889	.10861	94	6939.8	40.171	.17525
75	4417.9	25.566	.11156	95	7088.2	41.020	.17900
76	4536.5	26.253	.11456	96	7238.2	41.888	.18278
77	4656.6	26.948	.11759	97	7389.8	42.765	.18661
78	4778.4	27.653	.12067	98	7543.0	43.651	.19048
79	4901.7	28.366	.12378	99	7697.7	44.547	.19439
80	5026.5	29.089	.12693	100	7854.0	45.451	.19833

EXAMPLE 25. — The diagrams in Fig. 84 are from a 14 in. by 15 in. engine, with 2½ in. piston rod and with 8 per cent of clearance in each end, running at 225 r.p.m. on a steam pressure of 105 lb. by gage. The indicator had a spring of 60 lb. scale, or the pressure scale of the diagrams is 60 lb. per sq. in. to the inch of height. Analyze the steam distribution and calculate the indicated horse-power.

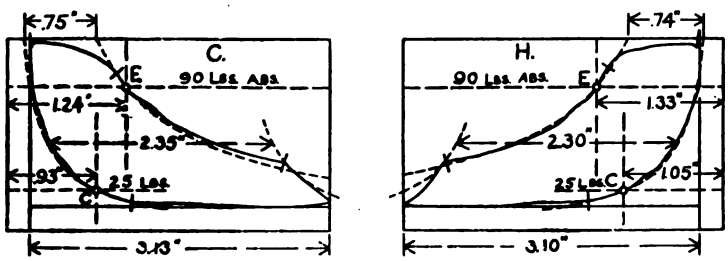

FIG. 84. — Diagrams from a High-speed Steam Engine.

The three important "events" in the steam distribution, cut-off, release, and exhaust closure or "compression," are marked on the diagrams by drawing short cross-lines, located by eye as closely as possible. Expressed in "apparent" measure — see § 19 (*d*) — or by their ratio of distance along the stroke line, and estimated from the beginning of forward stroke, or from the right end of the head diagram and from the left end of the crank diagram in Fig. 84, these events are as tabulated in Form 1.

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FORM 1. EVENTS IN STEAM DISTRIBUTION.

Diagram.	Cut-off.	Release.	Compression.
Head end	0.31	0.86	0.37
Crank end	0.29	0.85	0.36

Equilateral hyperbolas, $pv = C$, are drawn along both the expansion and the compression curves, being passed through the E and C points selected for the determination of indicated steam consumption, and laid out by the method of § 6 (*g*). The curves of the diagram conform very closely to the hyperbolas, and no comment is called for.

The calculation for horse-power is outlined in Form 2. Usually, it is sufficient and more convenient to place the data for m.e.p. upon the diagrams themselves; these data are combined as explained in Art. (*a*). Using the engine constants already found in Example 24, i.h.p. is found through the multiplication of the constant by r.p.m. and by m.e.p., according to Eq. (115). Thus for the head end,

$$H = 0.005831 \times 225 \times 48.35 = 63.43.$$

FORM 2. CALCULATION OF INDICATED HORSE-POWER.

Cylinder end.	Diagram area.	Diagram length.	Mean height.	M.e.p.	I.h.p.	
					Ends.	Total.
Head	2.50	3.10	0.806	48.35	63.43	124.40
Crank	2.40	3.13	0.796	47.70	60.97	

EXAMPLE 26. — Calculate the indicated horse-power of the 13 and 20 by 16-inch engine outlined in Fig. 79, with the diagrams in Fig. 78.

The operations are indicated and results given in Form 3. The m.e.p.'s were found with an averaging planimeter, so that the areas of the diagrams are not recorded.

FORM 3. INDICATED HORSE-POWER.

Cylinder end.	Piston area.	Constant *C*.	M.e.p.	I.h.p.		
				Ends.	Cylinders.	Total.
H. H.	132.7	0.005360	26.7	35.8	75.5	157.0
H. C.	130.3	0.005266	30.15	39.7		
L. H.	311.8	0.01260	14.22	44.8	81.5	
L. C.	309.3	0.01249	11.75	36.7		

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(e) INDICATED STEAM CONSUMPTION. — The data needed for calculating this quantity are outlined on Fig. 85, and the method of procedure will now be described. When the piston is at the position defined by the point E on the expansion curve, the space FE is filled with steam of the pressure p_e and specific volume (dry saturated) s_e. The volume FE or V_e is preferably expressed by its ratio to the nominal volume or

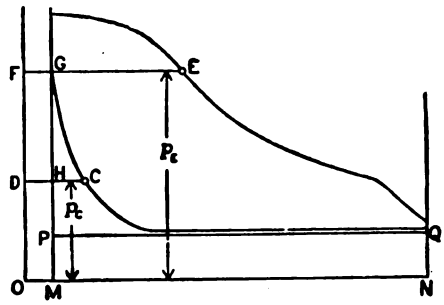

FIG. 85. — Data for Indicated Steam Consumption.

piston displacement V, which on the diagram is represented by the stroke line MN. The relations involved are

$$e = \frac{\text{FE}}{\text{MN}}, \qquad V_e = eV \text{ cu. ft.};$$

and the weight of steam (vapor alone) present in the cylinder at E is

$$w_e = \frac{eV}{s_e} = \frac{e}{s_e} \times V.$$

This measures the total steam; in order to deduct the clearance steam, similar expressions must be written for a point C on the compression curve, namely,

$$c = \frac{\text{DC}}{\text{MN}}, \qquad V_c = cV, \qquad w_c = \frac{c}{s_c} \times V.$$

Now for the net or working steam apparently used per revolution the value in pounds is

$$w = w_e - w_c = V\left(\frac{e}{s_e} - \frac{c}{s_c}\right). \quad . \quad . \quad . \quad . \quad . \quad (117)$$

Letting K stand for the expression in parentheses, this formula becomes

$$w = KV; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (118)$$

and since $K = w \div V$, it is evident that the quantity K is the indicated steam consumption per cubic foot of piston displacement.

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The simplest case in the determination of K occurs when the points E and C can be taken on the same horizontal line; then

$$K = \frac{e - c}{s_\theta}, \text{ and } (e - c) = \frac{\text{CE}}{\text{MN}}: \quad . \quad . \quad . \quad . \quad (119)$$

but it is better not to take C much above the beginning of compression.

(*f*) I. S. C. PER HOUR AND PER HORSE-POWER-HOUR. — Now the piston makes $60N$ outstrokes in one hour, N being the number of revolutions per minute, or the r.p.m.; then the i.s.c. in pounds per hour is

$$I_{\text{H}} = 60\,KNV. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (120)$$

This is for one end of the cylinder, or for one diagram: a similar value would be found for the other end, and the sum of the two would give the total steam shown by the indicator.

If we let V_{H} stand for the total piston displacement in cubic feet per hour in both directions, so that

$$V_{\text{H}} = 60\,N\,(V_1 + V_2), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (121)$$

and find a mean value of K from the two diagrams, then the simplest way to get the i.s.c. per hour, total, is to substitute in

$$I_{\text{H}} = K_{\text{m}} V_{\text{H}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (122)$$

In terms of stroke and piston area,

$$V = \frac{AS}{1728} \text{ and } I_{\text{H}} = K\frac{ASN}{28.8},$$

from Eq. (120); dividing this by the formula for i.h.p. in Eq. (115), we get

$$I = \frac{I_{\text{H}}}{H} = \frac{13750}{p_{\text{m}}} K, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (123)$$

where I_{H} stands for i.s.c. per h.p.h.

The method of Eq. (123) is generally used when isolated diagrams are worked up, while that of Eqs. (120) and (122) is preferable for long tests with fairly uniform conditions, especially if there is a parallel determination of actual steam consumption. Note that in Eq. (123) all the dimensions of the engine have been canceled out, so that the quantity I is dependent wholly upon the form of the indicator diagram. Note further that when two values of I have been found from a pair of diagrams, their *mean*, not their sum, is to be taken to get a result for the whole engine; because each value is the ratio of the steam passing through one end to the work done by that steam.

The point E may be located anywhere along the expansion curve, but is usually taken either just after cut-off or just before release. On

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account of reëvaporation during expansion, the latter position will usually give a larger i.s.c. than the former.

EXAMPLE 27. — Calculate the indicated steam consumption from the diagrams in Fig. 84, Example 25.

The dimensions necessary for getting K are all marked on the figure. For the head end, $e = 1.33 \div 3.10 = 0.429$, and with $p_e = 90$ lb., $s_e = 4.960$. The whole calculation is set forth in Form 4.

FORM 4. CALCULATION OF INDICATED STEAM CONSUMPTION.

Cylinder end.	e	c	$\frac{s_e}{s_c}$	$\frac{e}{s_e}$	$\frac{c}{s_e}$
Head	0.429	0.339	4.960	0.0865	0.0208
Crank	0.396	0.297	16.29	0.0798	0.0182

Cylinder end.	K	$\frac{13750}{\text{m.e.p.}}$	I.s.c.	
			Ends.	Mean.
Head	0.0657	284.5	18.70	18.23
Crank	0.0616	288.0	17.75	

Here Eq. (123) is used, after K has been found, and the result is in pounds per horse-power-hour. If total indicated steam per hour is desired, results in Forms 2 and 4 can be combined as follows:

$I_H = 124.4$ (i.h.p.) $\times$ 18.23 (i.s.c.) = 2268 lb. Or, by the method of Eq. (121), after finding the K's take volumes from Example 24 and note that with 225 r.p.m. the revolutions per hour are 13,500; then

		Head.	Crank.
Cylinder volume,	V	= 1.336	1.301 cu. ft.
Piston displacement,	V_H	= 18036	17564 per hour.

And $I_H = \begin{pmatrix}\text{Head} & 18040 \times 0.0657 = 1185 \\ \text{Crank} & 17560 \times 0.0616 = 1080\end{pmatrix} = 2265$ lb.

(g) I. S. C. FROM COMPOUND-ENGINE DIAGRAMS. — The indicated steam coefficient K is calculated from any one diagram, or for any cylinder end, in the same manner as for a simple engine. It is then rather the more direct procedure, in idea at least, to get the total i.s.c. or I_H by the method of Eq. (122), and divide by total i.h.p. or H. The calculation of the steam per horse-power-hour I directly from the diagram by the method of Eq. (123), is complicated by the fact that, in the multiple-expansion engine, the steam which is metered in one cylinder does work in two or more. In order to derive Eq. (123) through the

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division of I_H by H, it is necessary that both these quantities be expressed in terms of the same cylinder dimensions; and we have occasion to use the idea of reduced mean pressure, already developed in § 20 (*b*). Suppose for instance, that K has been found from a high-pressure diagram: in this cylinder end, the mean effective pressure p_1 acts upon area A_1; and the same steam then proceeds to do in the large cylinder the work represented by p_2 as m.e.p. on the area A_2. Now the latter amount of work would be done upon the first piston by a reduced m.e.p. of the value,

$$p_r = p_2 \frac{A_2}{A_1} = r p_2, \quad \ldots \ldots \ldots \quad (124)$$

acting on the area A_1. In this formula, r stands, in general, for the ratio of the area upon which the pressure is actual to that to which it is to be reduced. Finally, a total m.e.p. of the value $p_m = p_1 + p_r$, introduced into Eq. (115), will give the total i.h.p. developed by the steam, in terms of the dimensions of the high-pressure cylinder; which dimensions will then cancel out in the division leading to Eq. (123).

With a three- or four-stage compound, there would be two or three reduced m.e.p.'s to be added to the actual m.e.p. in the cylinder end for which the i.s.c. was being found.

EXAMPLE 28. — Calculate indicated steam consumption from the compound-engine diagrams in Fig. 78, using horse-power results from Example 26 as needed.

The measurements required for this problem are marked on the diagrams as reproduced in Figs. 80 and 81. From these are calculated the values of K for each cylinder end, in Form 5, the E points being taken near high-pressure cut-off and low-pressure release, or near the beginning and the end of the total expansion. Then the i.s.c. is worked out, by the first method above, in Form 6; the characteristic operation is the division of I_H by H, or, for the high-pressure cylinder, of 2016 lb. of indicated steam by 157.0 i.h.p. Note how much larger I is for the low than for the high cylinder; as remarked in § 20 (*h*), this is abnormal and is due to leakage through or past the high-pressure cylinder.

FORM 5. CALCULATION OF K, FOR I.S.C.

Cylinder end.	e	s_e	c	s_c	$\frac{e}{s_e} - \frac{c}{s_c} = K.$
H. H.....	0.438	4.894	0.316	8.50	0.0895−0.0372=0.0523
H. C.....	0.433	4.894	0.258	8.50	0.0885−0.0304=0.0581
L. H.....	0.977	26.78	0.244	38.40	0.0365−0.0064=0.0301
L. C.....	0.714	26.78	0.160	38.40	0.0267−0.0042=0.0225

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FORM 6. I.S.C. PER HOUR AND PER H.P.H.

Cylinder end	Cylinder volume.	$60NV$ or V_H	K	I_H Ends.	I_H Totals.	I.
H. H.	1.229	18,435	0.0523	964	2,016	12.84
H. C.	1.207	18,105	0.0581	1,052		
L. H.	2.887	43,305	0.0301	1,304	2,269	14.45
L. C.	2.860	42,900	0.0225	965		

The calculation of I from K by Eq. (123), with the use of reduced or referred m.e.p. involving the relation expressed in Eq. (124), is set forth in Form 7. The actual m.e.p.'s are first entered under p_n, then each is multiplied by the volume ratio just ahead of it, and the result entered under p_r, in the line belonging to the cylinder end in which is carried out the other part of the cycle. Thus for high head (H.H.), $26.70 \times 0.429 = 11.47$, and this is entered opposite L.C.; and so on. The rest of the calculation is obvious. The difference between the two values of mean I in the low-pressure cylinder — 14.37 as found here and 14.45 in Form 6 — is due to the fact that the mean of the ratios of partial quantities is not the same as the ratio of the sums: the method of Form 6 is inherently the more correct.

FORM 7. INDICATED STEAM PER HORSE-POWER HOUR DIRECTLY.

Cylinder end.	Volume ratio r.	Total M.e.p. $p_n + p_r = p_m$	$\frac{13750}{p_m}$	K.	I.s.c. Ends.	I.s.c. Means.
H. H.	$\frac{\text{H. H.}}{\text{L. C.}} = 0.429$	$26.70 + (27.36) = 54.06$	254.1	0.0523	13.29	12.87
H. C.	$\frac{\text{H. C.}}{\text{L. H.}} = 0.418$	$30.15 + (34.09) = 64.15$	214.1	0.0581	12.44	
L. H.	$\frac{\text{L. H.}}{\text{H. C.}} = 2.39$	$14.22 + (12.60) = 26.82$	512.4	0.0301	15.42	14.37
L. C.	$\frac{\text{L. C.}}{\text{H. H.}} = 2.33$	$11.75 + (11.47) = 23.22$	591.8	0.0225	13.31	

(*h*) MEASUREMENT OF STEAM CONSUMED. — As intimated in § 19 (*n*), the only way to find the amount of steam actually used by an engine is to weigh or measure that steam as water. The feed-water method is available when the output of one or more boilers can be wholly devoted to the engine under test, or, when, at least, steam is deflected only to small auxiliaries like the feed and condenser pumps, the exhaust from which can be condensed and weighed. This scheme

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is liable to the error caused by an insufficiently accurate determination or equalization of the amount of water in the boiler at the beginning and end of the test. The surface-condenser method gives the surer results, but calls for a supply and arrangement of apparatus which often is not available, except in a laboratory or special testing plant. In the more complicated engines there are a number of minor steam quantities to be measured, such as the amounts of water drained from the several jackets and receivers.

(*i*) CURVES OF STEAM CONSUMPTION. — In Fig. 86 are shown the two curves commonly used for representing the relation between steam consumed and power developed. Here indicated horse-power is the base, but effective power will serve equally well. Curve AA shows total consumption per hour, curve BB the consumption per horse-power-hour. The diagrams are most useful and effective when the amount of steam admitted per cycle is the principal variable, the limiting pressures and the speed being kept nearly constant as the load changes: some of the irregularities in Fig. 86, or the failures of the points to conform to smooth curves, are due to the variations in speed.

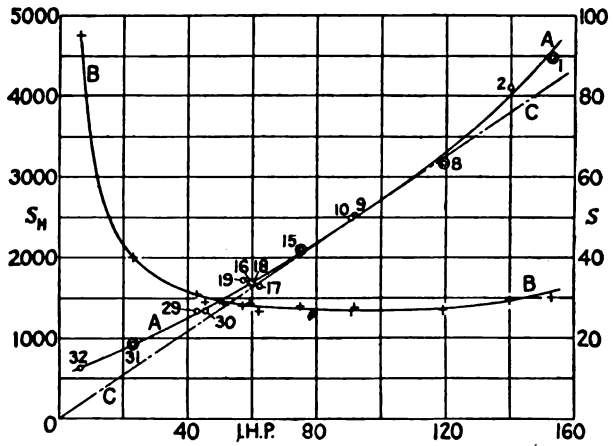

FIG. 86. — Curves of Steam Consumption, representing tests of a 17 by 30 inch air compressor by Professors Denton and Jacobus. Trans. A.S.M.E., Vol. 10, 1889, page 722. Controlling conditions: steam pressure, 89 to 92 lb. by gage; exhaust (into surface condenser) at pressure of atmosphere; speed about 60 r.p.m., except that tests 1, 8, and 15 are at higher speeds — see Fig. 90. The test numbers are from the original report.

The curves in this figure are typical in form, well representing the characteristic behavior of the steam engine. As shown by curve BB, the steam S per horse-power-hour has a minimum which marks the condition of best economy (here at about 100 horse-power), and rises

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from this in either direction, slowly at first, then at an increasing rate. The degree of underloading is sufficient in this series of tests to let excessive cylinder-wall action make S run very high. With heavier overloading there would be a similar marked rise of curve BB at its right-hand end, showing the loss due to an increasing lack of expansion after cut-off. Indicator diagrams from this engine are given in Fig. 90. and the effect of changes in condition will be more fully discussed presently.

If the steam per horse-power-hour were constant over the whole range of loading, the curve AA would become a straight line, running to the origin of coördinates at O. Drawing then the line CC, we can quite readily see from curve AA alone the manner in which extreme conditions make the steam rate rise above its best (lowest) value.

(*j*) THE DIAGRAM OF SPECIFIC STEAM CONSUMPTION. — In order to eliminate speed and size of engine as factors in steam quantity — which is desirable for certain purposes — the scheme of Fig. 87 has been developed. This follows the idea of curve AA, Fig. 86, in representing an absolute rather than a relative quantity; but, incidentally, the steam per horse-power-hour is also shown. The ordinate of the diagram is the steam consumed per cubic foot of piston displacement, the same kind of quantity as K in Arts. (*e*) and (*f*). To get it, the steam per hour, S_H, is divided by $V_H = 60\,NV$: working for the whole cylinder, rather than for one end, V is taken as the double volume, or the volume displaced by the two piston faces in the two strokes which make up one revolution. With speed and size thus divided out, the rational analogue to horse-power is the mean effective pressure, and this is taken as the base in Fig. 87.

The letter K will now be used as a general symbol for weight of steam per cubic foot of piston displacement, with subscripts to designate particular values — note particularly that the various K quantities are in direct proportion to steam per revolution, being the latter divided by the nominal cylinder volume. The relation expressed in Eq. (123) holds between any K and the corresponding steam per horse-power-hour. Writing the equation as

$$S = \frac{13750}{p_m} K, \quad \ldots\ldots\ldots \quad (125)$$

it appears that for any fixed value of S there is a straight-line relation between K and p_m. The inclined lines radiating from zero on Fig. 87, marked with numbers which are values of S, come from this formula. The points along the curve SS show steam consumed, the same quantity as is plotted in Fig. 86: referred to the scale at the left, they give

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values of K, while on the scale formed by the inclined lines values of S can be read off. The lower points show the steam condensed by the cylinder, and will be considered in the next article.

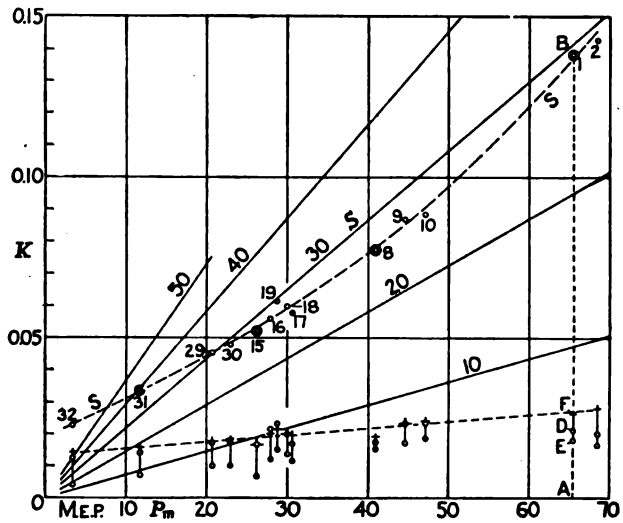

FIG. 87. — Specific Steam Consumption: weight of steam per cubic foot of piston displacement, plotted on mean effective pressure as base. Same engine and tests as in Fig. 86. Tests 1, 8, 15, and 31, indicator diagrams in Fig. 90.

EXAMPLE 29. — Engine 17 by 29½ in., rods $2\frac{2}{16}$ in. head end, $2\frac{1}{4}$ in. crank end. In test No. 1, m.e.p. = 65.42, r.p.m. = 70.38, i.h.p. = 153.0, steam per hour S_H = 4571 lb., steam per horse-power-hour S = 29.88 lb. Find K as plotted on Fig. 87, also fix the S-constant line for 30 pounds.

First get V with the help of Table 9, as follows:

		Head.	Crank.
For the full piston,	V_0 =	1.3135	1.3135
For the rod,	V_0 =	0.0299	0.0376
Net value of V_0,		1.2836	1.2759

Adding and multiplying by $S/10$ (this S being "stroke," not "steam") we get, for the double displacement,

$$V = 2.5595 \times 2.975 = 7.615 \text{ cu. ft.};$$

then $V_H = 60 \times 7.615 \times N = 456.9\,N$ cu. ft., this number 60 V being an important engine constant in the present connection.

The value of K is now, very simply,

$$K = 4571 \div (70.38 \times 456.9)$$
$$= 4571 \div 32157 = 0.1422 \text{ lb. per cu. ft.}$$

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To locate the steam line for $S = 30$, one point is enough: let it be on the ordinate line for $p_m = 60$, then from Eq. (125)

$$K = \frac{Sp_m}{13750} = \frac{30 \times 60}{13750} = 0.1309 \text{ lb.}$$

(*k*) VARIOUS STEAM QUANTITIES. — Consider the various lengths along the ordinate AB in Fig. 87; the quantities represented are as follows:

AB or K is the actual steam consumed by the engine, the only quantity plotted in Fig. 86.

BD or K_{ic} is the indicated steam at cut-off, the same as plain K in Eqs. (118) to (123); then

AD or K_{mc} is the "missing quantity," or the amount of steam condensed by the cylinder walls, as determined at cut-off.

Similarly, BE = K_{ir} and AE = K_{mr} show indicated steam and condensed steam at release, and ED is the amount of reëvaporation. Sometimes the last is a negative quantity, E falling above D; when this condition exists it is indicated by placing an arrowhead, pointing upward, on the line DE, or by placing a small arrow beside the points, as in Fig. 97.

Finally, AF or K_{mf} is the missing quantity at cut-off, computed by an empirical formula which will be presented and discussed in the next section. Without the second subscript, K_i and K_m will be used as general symbols for indicated steam and for condensed or missing steam.

In further illustration of the possibilities of the specific diagram, Fig. 88 is given at this point. Here a portion of the steam coming to to the engine is diverted into the jackets and reheater — the latter consisting of a nest of pipes or tubes placed within the receiver, for the purpose of drying and perhaps slightly superheating the steam going to the low-pressure cylinder. Both the jackets and the reheater are supplied with steam of full boiler pressure. Now, on the diagram,

AB = K_t = total steam consumed;
AC = K = steam consumed in cylinder;
BC = K_j = jacket steam.

Note that the base in this figure is the m.e.p. reduced to the low-pressure cylinder of the compound engine. A new feature here introduced is the horse-power scale at the top of the diagram, laid out for the average speed of running, which is combined, of course, with the m.e.p. as scaled at the bottom. The main purpose of the diagram is the representation and study of steam quantity, especially of the secondary quantities like K_m and K_j which are involved in the action of the cylinder walls. From this point of view an approximate idea of

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the power of the engine is enough, and when the speed varies only as permitted by the governor, the horse-power scale gives the load quite effectively. In the engine whose performance is represented by Fig. 88, the speed varied from 120.6 to 122.7 over the range of loading, or by 1.7 per cent.

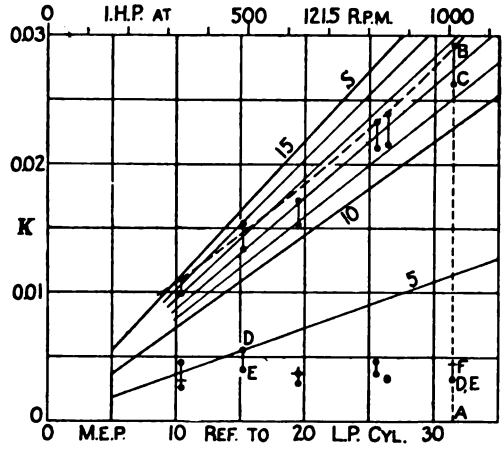

FIG. 88. — Tests of a 20 and 40 by 42 inch Corliss Cross-compound Engine, with steam jackets and a reheating receiver. D. S. Jacobus, Trans. A. S. M. E., Vol. 24, 1903, page 1274. No. 26 in Table 13, page 268.

(*l*) THE UNIT STEAM DIAGRAM. — Having the specific steam consumption K, it is an easy matter to get the length of volume base for a steam diagram to represent the performance of one pound of steam. If the engine consumes K lb. of steam per cubic foot of piston displacement, the displacement volume corresponding to a consumption of one pound of steam is evidently $1 \div K$. Thus in Example 29, K is found to be 0.1422 lb. per cu. ft.; then the base length MN (representing the stroke line, not including clearance length) is $1 \div 0.1422 =$ 7.03 cu. ft., which is used in Fig. 90.

In the case of Fig. 82, a slightly different procedure is more convenient. As the engine had not been tested for steam rate, it was necessary to assume (for illustrative purposes only) a probable value of steam consumption. This was taken as 17.5 lb. per h.p.h., which is probably too low, in view of the strong indications of leakage. With 157.0 i.h.p. (from Example 26), the hourly rate is $W_H = 157 \times 17.5 =$ 2748 lb. of steam; and at 250 r.p.m. the number of cycles is 30,000 per hour. Then the consumption of steam is $2748 \div 30{,}000 = 0.0916$ lb. per cycle. If now the diagrams with actual engine volumes represent the performance of 0.0916 lb. of steam, the unit diagram must be larger in the ratio of 1.00 to 0.0916; or, the volume measures from Figs. 80 and

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81 must be multiplied by 10.92 before they are laid off to scale in Fig. 82. To get the limiting volumes OM and MN in the usual diagram like Fig. 80 (full-line), the actual volumes of displacement and clearance would be multiplied by this factor.

§ 22. Effect of the Cylinder Walls

(*a*) General Conditions of Knowledge. — A great deal of information has been accumulated as to the resultant effect of the cylinder walls upon the performance of the steam in the engine, but in regard to the intimate detail of the thermal interchanges we have very little exact and reliable knowledge. With good indicator diagrams and a careful measurement of steam consumed, it is a comparatively easy matter to determine the net effect of the wall action, as represented by the magnitude and manner of variation of the missing quantity, or of the difference between actual and indicated steam: the methods required have just been developed. It is not at all difficult, although more troublesome, to go a step farther, and from these same data deduce the amounts of heat transferred back and forth, between steam and metal. The problem in thermal physics, for the rational, synthetic solution of which both method and data are almost completely lacking, would be stated about as follows:

Given an engine with a known volume of cylinder and area of internal surfaces, working through a certain range of pressure and of steam temperature, and with the time and timing of the cycle fixed by the general speed of running and by the action of the valve or valves; with these data, find how much the temperature of the metal will vary, how much heat will be interchanged between steam and metal, and what amount of initial condensation will occur.

In the present section, the question of cylinder losses is quite fully studied, by the empirical method of comparing and combining results as to the missing steam; that which follows is devoted to the effect of compression upon economy. In § 24 the methods of thermal analysis are described and illustrated, while § 25 summarizes the comparatively small amount of data available in the direction of a solution of problems like that just stated.

(*b*) A Formula for Cylinder Condensation. — The writer has devised a formula for the proportion of initial condensation, which harmonizes the variant data about as well as can be expected, and has a wide range of application. The formula is

$$m = \frac{C}{\sqrt[3]{N}} \sqrt{\frac{sT}{pe}}; \quad \ldots\ldots\ldots \quad (126)$$

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in which

m is the fraction of initial condensation, or the ratio of the missing quantity to the total steam; commonly, and most conveniently this base is the steam entering the cylinder; so that in terms of Fig. 87 and the definitions in § 21 (k), m is equal to K_{mc}/K. But when there is a great deal of compression, or a large relative weight of compressed steam, m is to be considered as the ratio of the missing quantity to the total weight of substance present during expansion.

C is a constant or coefficient, which must be changed to some extent with the type of engine; a good mean value is 0.27, and the ordinary range is from 0.25 to 0.30.

N is the speed of running, in revolutions per minute.

s is a constant obtained (for any engine) by dividing the surface of the nominal cylinder in square feet by its volume in cubic feet; this factor is an approximate inverse measure of the size of the cylinder.

T represents the range of steam temperature within the cylinder during the cycle, from the highest pressure p_1 to the lowest pressure p_2, these pressures being taken from the indicator diagram. It is more fully defined on page 178.

p is the absolute pressure at the point E, on the expansion curve at or just after cut-off, for which the indicated steam consumption is found.

e representing the ratio of cut-off or the amount of expansion in the cylinder, has the same meaning as in the calculation of indicated steam consumption; it is the total volume out to E divided by the nominal cylinder volume V.

The formula is intended to give the missing quantity at cut-off, in an engine which is tight enough to show but an inappreciable amount of leakage at valves and piston, tested when standing still, in the usual fashion — see Art. (p). The engine is supposed to receive ordinary, "commercially dry" steam, which generally shows from 0.5 to 1.5 per cent of moisture when tested with the throttling calorimeter. The formula is not applicable to the lower cylinders of multiple-expansion engines, except in the few cases where the water in the exhaust from the high cylinder is allowed to separate out in the comparative quietude of a large receiver and is drained away, leaving dry steam to enter the lower cylinder. It does not cover the use of superheated steam, nor does it apply when steam jackets are in action.

(c) BASIS OF THE CONDENSATION FORMULA. — Equation (126) is rational as to the elements involved, but empirical as to the amount of

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influence which each exerts; it is justified, of course, only to the degree in which its results agree with those of reliable experiments. Three major considerations have been incorporated into the formula, namely,

1. The time of the whole cycle, proportional to $1/N$.

2. The amount of metal surface exposed per unit of steam affected, taken as roughly proportional to s/pe.

3. The range of steam temperature within the cylinder, represented by T.

(1) The strength of the time influence is purely empirical; in close analysis it appears that the exponent of the divisor N ranges from 0.3 to 0.4, but $\frac{1}{3}$, or the cube root, is found to be quite a satisfactory mean value.

(2) It would be logical to use the whole clearance surface — cylinder head plus piston face plus port surfaces — together with the portion of the cylinder barrel out to cut-off, and compare this with the total amount of steam admitted or present, including that condensed. As regards the cylinder, this would require information which usually is not available; while on the side of steam quantity most undesirable complication would be introduced into what can be, at best, but an approximate estimate of the missing quantity. The actual procedure is as follows:

The specific volume of steam is taken to be nearly proportional to $1/p$, so that pe represents the indicated steam per cubic foot of displacement, or K_{ic}. As defined above, s is the nominal surface per cubic foot of displacement; then s/pe is at least roughly proportional to the surface per pound of steam. The useful assumption is not so much that s represents actual clearance surface as that it varies in about the same manner with size of cylinder. Letting D be diameter and S stroke of piston, both in inches, and using for the total surface two circles of diameter D plus the cylindrical barrel of length S, we have,

$$s = \frac{\pi}{144} D\left(\frac{D}{2} + S\right) \div \frac{\pi}{4} \frac{D^2 S}{1728}$$

$$= \left(\frac{D}{2} + S\right) \frac{4 \times 12}{DS} = \frac{12}{D}\left(2\frac{D}{S} + 4\right) . \quad . \quad . \quad . \quad (127)$$

(3) It is apparent that the amount of heat interchange between steam and cylinder walls will depend upon the range of temperature in the cylinder. But it was found that the very large increase in range caused by dropping to condenser temperatures exerted too great an influence in the formula; while in the first cylinder of a multiple-expansion engine the range was too small to account for the condensation. To use the range of pressure would give an error in just the opposite

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direction, making the "moisture" figure out too small with condensing engines, too large with compounds.

To get around this difficulty, at the same time avoiding any complex mathematical expression in terms of temperature, the artificial function T was laid out by trial; this is shown in Fig. 89, plotted on p as a base, with the temperature curve (from Table I) dotted in for comparison. To get T for the formula, look up T_1 and T_2, corresponding to the highest and lowest pressures in the cylinder, then

$$T = T_1 - T_2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (128)$$

Do not confuse this T with the absolute temperature.

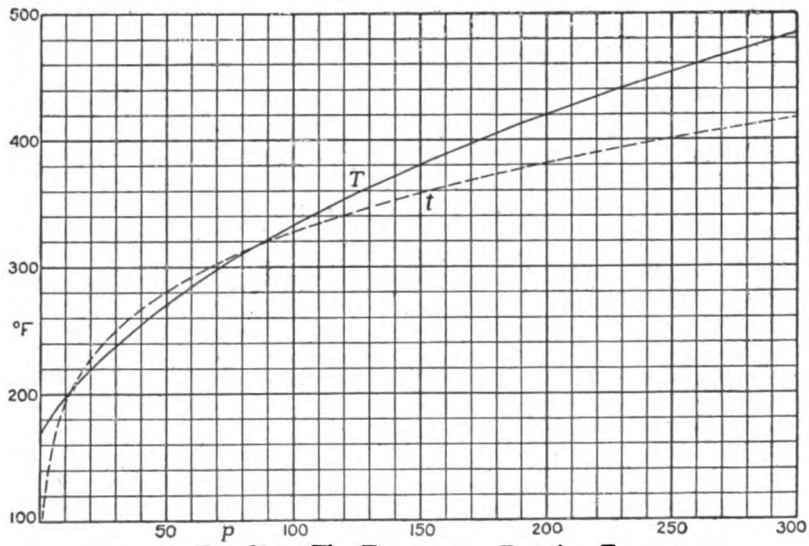

FIG. 89. — The Temperature Function T.

EXAMPLE 30. — Same engine as in Example 29, 17 by 30, mean clearance 0.071, double slide valve (Meyer gear — see Fig. 252). In Test No. 10, $p_1 = 108.1$, $p_2 = 16$, both absolute, $N = 58.0$, cut-off is at 0.313 of stroke and at 80.5 lb. abs. Compute the probable condensation ratio m.

From a handbook table of powers and roots, or from the slide rule,

$$\sqrt[3]{N} = \sqrt[3]{58.0} = 3.87.$$

By Eq. (127),

$$s = \frac{12}{17}\left(\frac{34}{30} + 4\right) = \frac{12 \times 5.13}{17} = 3.62,$$

$$0.27\sqrt{s} = 0.514.$$

From Table 10,

$$p_1 = 108.1, \quad T_1 = 341.1$$
$$p_2 = 16, \quad T_2 = 212.0 \quad T = 129.1.$$

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From the cut-off and clearance as given.

$$e = 0.313 + 0.071 = 0.384, \qquad pe = 30.9.$$

By Eq. (126),

$$m = \frac{0.514}{3.87}\sqrt{\frac{129.1}{30.9}} = 0.271.$$

According to the report of the tests — see reference under Fig. 86 — the fraction of steam not accounted for by the indicator is, at cut-off, $m_c = 0.264$, at release, $m_r = 0.211$, so that in this case the formula shows most excellent agreement with the test. Use of the slide rule is especially appropriate and convenient in the above calculation.

TABLE 10. VALUES OF T FOR EQUATION (126).

p	T	p	T	p	T	p	T
0	170	45	262	115	348	185	409
1	175	50	269.5	120	353	190	413
2	179	55	277	125	358	195	416.5
3	183	60	284	130	362.5	200	420
4	186	65	291	135	367	210	427
6	191	70	297.5	140	371.5	220	434
8	196	75	304	145	376	230	441
10	200	80	310	150	380.5	240	447.5
15	210	85	316	155	385	250	454
20	220	90	321.5	160	389	260	460.5
25	229	95	327	165	393	270	467
30	238	100	332.5	170	397	280	473
35	246	105	338	175	401	290	479
40	254	110	343	180	405	300	485

p is absolute pressure in pounds per square inch.
T is the temperative function in degrees fahrenheit.

(*d*) VARIOUS ENGINE TESTS. — As already remarked, an empirical formula like Eq. (126) is justified only in so far as it fits the facts, or agrees with experimental data. A number of important sets of engine tests will now be diagrammed and briefly described, with the double purpose of showing actual performance and of trying out the condensation formula. The method of Fig. 87 is used, the missing steam being represented, not by the ratio m, but in actual magnitude as K_m or mK. Conclusions or summarized statements will be found in Arts. (*g*), (*j*), (*l*), (*n*), and (*o*), while leakage is discussed in Art. (*p*). A further presentation of data, in condensed numerical form, is made in Table 13, § 27 (*j*).

It must be fully understood that the probability of a definite and closely-acting law, relating the amount of cylinder condensation to the controlling conditions, is rather low. Nevertheless, the development of a fairly good approximation is of great advantage and utility. The

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applications of the formula are two: it may be used to predict or estimate the probable actual consumption from the data of a simple indicator test of an engine; and it will serve as a standard of comparison in steam-rate tests of working engines, upon which to base an estimate as to how much of the missing quantity is due to leakage. Further, in the discussion of test-series where there are changes in other than the principal variable, the use of K_{mf} from Eq. (126) as a standard of comparison for observed K_{mo} practically eliminates the effect of these secondary changes.

In any particular engine, the controlling conditions are the two limiting pressures, the ratio of cut-off, and the speed. Variation of cut-off with load is the commonest change, in power service at least, and that type of variation will be considered first.

(*e*) VARIATION WITH CUT-OFF. — Fig. 87 is a very good example in this connection, to be accompanied by the indicator diagrams in Fig. 90. On the latter figure, the speeds are given under N and the i.h.p.'s

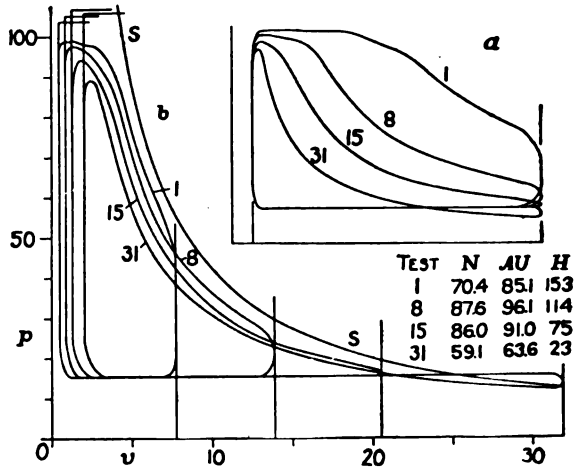

FIG. 90. — Steam Diagrams from a 17 by 30 inch Noncondensing Engine, mean clearance 0.071, tests plotted in Figs. 86 and 87. Mean indicator diagrams at *a*, unit diagrams at *b*: in regard to layout of the latter, see § 21 (*l*).

under H; while AU is the indicated work output per pound of steam, or the net external work of the steam, expressed in heat units. The thermal value of one horse-power-hour being 2545 B.t.u., and the engine consuming S lb. of steam per h.p.h., the output per pound is,

$$AU = 2545 \div S. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (129)$$

The curve SS is the saturation line, or the curve of constant steam weight for one pound of steam. The distance between this line and

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the expansion curve of any diagram does not exactly represent the missing steam, because the steam back of that curve includes the clearance steam, in addition to the pound of working steam. To get a true measure of quality, it would be necessary to produce the compression curves upward — in idea, as curves of constant steam weight, but, with such proportions as here exist, about equally well as hyperbolas — then divide the intercept between expansion and compression curves by the distance from the vertical axis to SS. Compare the compound-engine diagrams in § 27.

In Fig. 87, the fraction m ranges from 0.14 in test 2 to 0.55 in test 32. Roughly, the quantity K_{mc} is nearly a constant, falling off slightly toward both ends of the range of m.e.p. Fig. 90 makes clearly evident the way in which the increasing relative size of the missing quantity cuts down the effective work area, soon neutralizing the gain from longer expansion. Test 8 shows about the condition of best efficiency, as appears also from its location on Fig. 87.

There is a strong element of historical interest in the tests set forth by Figs. 91 and 92, since these experiments were among the first clearly

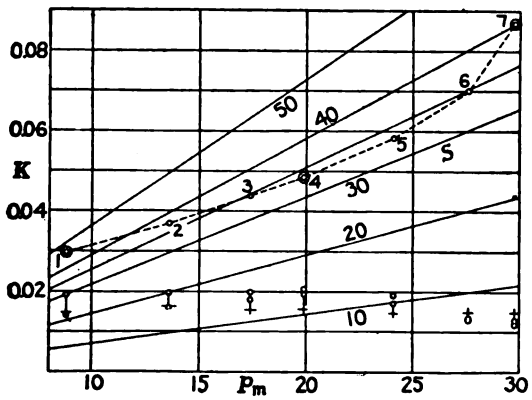

Fig. 91. — Tests of 36 by 96 in. Low-pressure Condensing Engine, U. S. S. *Michigan*, made in 1860. Isherwood, *Researches in Steam Engineering*, Vol. I, pages 91 to 120. Paddle-wheel revenue cutter, duplex engine, one cylinder used in tests, vessel moored to dock. Boiler pressure 20 lb. by gage, speed 11 to 21 r.p.m. I. s. c. recomputed with present steam tables. For scheme of diagram, see Fig. 87.

to establish the fact and the harmful effect of cylinder condensation, showing that there is an early limit to the amount of expansion that can economically be carried out in one cylinder. In Fig. 92, the shrinkage of volume is more striking than in Fig. 90. Here hyperbolas are drawn along the expansion curves, on the two diagrams with earlier

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cut-offs, also the very short compression curves are produced upward. The saturation curve is for one pound of steam, as in Fig. 90. Fig. 91 shows that K_{mc} holds nearly constant for tests 1 to 5, then falls off quite

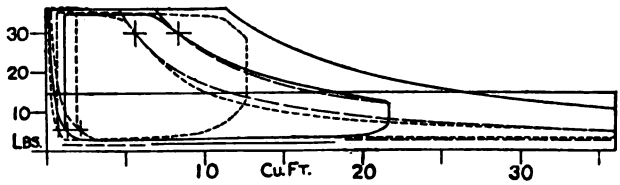

FIG. 92. — Unit Diagrams from the *Michigan's* Engine, Tests 1, 4, and 7. Clearance, 0.058; range of cut-off, 0.09 of stroke at 33 lb. abs. to 0.92 at 32 lb.

sharply. The formula points agree fairly well with a horizontal straight line, as against a slightly inclined line in Fig. 87. To make K_{mf} equal K_{mc} on an average of the seven tests, constant C in Eq. (126) would have to be raised to 0.30; while for average conformity in tests 1 to 5, the coefficient must be made a little over 0.34.

(*f*) OTHER SERIES OF TESTS in which cut-off is a principal variable are diagrammed in Figs. 93 to 97. To supplement the showing of Fig.

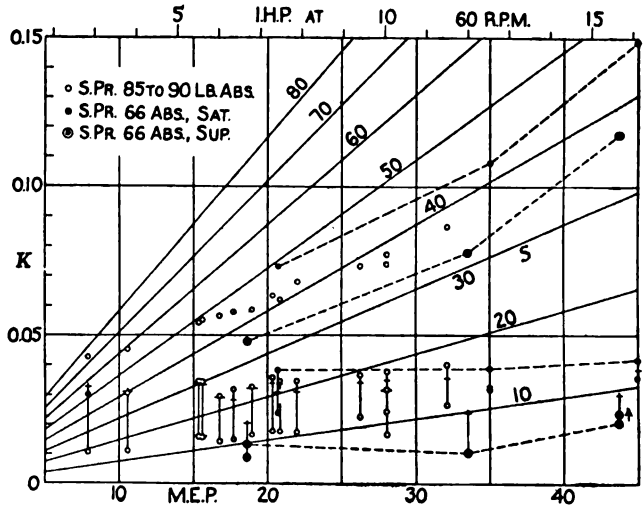

FIG. 93. — Tests of an 8 by 24 in. Noncondensing Corliss Engine, Massachusetts Institute of Technology, about 1884; Peabody's *Thermodynamics*, earlier editions, also Trans. A. S. M. E., Vol. 7. Clearance, 0.04; range of cut-off in main group of tests, from 0.03 of stroke at 74 lb. abs. to 0.24 at 72 lb. For scheme of diagram, see Fig. 87.

87, the performance of a very small Corliss engine is given in Fig. 93. The ratio m_c runs very high, ranging from 0.43 to 0.70, yet the formula value m_f keeps close to it. The minor groups, of three tests each, show

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the gain from superheating in this engine. The amount of superheat varied from 103 deg. fahr. at the heaviest load to 175 deg. at the lightest. With so much room for saving, superheating effects a marked reduction in the steam used.

As examples of large, low-pressure engines at high speeds, the performance of the low-pressure cylinders of a group of compound engines is given in Fig. 94. By draining the receiver and weighing the water thus drained out, dry steam is supplied to the low cylinder in known

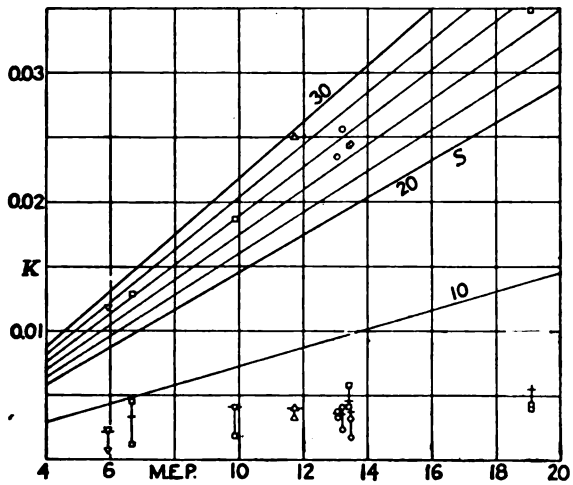

FIG. 94. — Large Low-pressure Condensing Engines; low-pressure cylinders of engines in the Marks tests, Fig. 97. Range of size, from 38 by 42 in. at 120 r.p.m. to 60 by 56 in. at 100 r.p.m. Initial pressure (within cylinder), 16 to 32 lb. abs., exhaust pressure 1.7 to 3.4 lb. abs. Range of cut-off, 0.14 to 0.47 of stroke.

amount, so that the latter can be treated as a simple engine, with its own m.e.p. and steam consumption per h.p.h. Of the full set of tests shown in Fig. 97, only those are here plotted in which the jackets and reheater were out of action. Except for engine B, the most erratic in Fig. 97 also, there is excellent agreement between formula and experiment. In these engines, K_m is less at light loads because the initial pressure (the receiver pressure in the whole engine) falls as the load is less and the high-pressure cut-off is earlier, so that the temperature range in the big cylinder is less.

The next example, Fig. 95, is complicated by the addition of tests made with the jacket, which are plotted as in Fig. 88. Unjacketed, K_{mf} and K_{mc} agree quite well. With the jackets (on heads and sides), it appears that at late cut-offs the saving in condensation within the

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cylinder is neutralized by the condensation in the jacket; but with earlier cut-off and a larger margin of possible saving, the jacket has a useful effect.

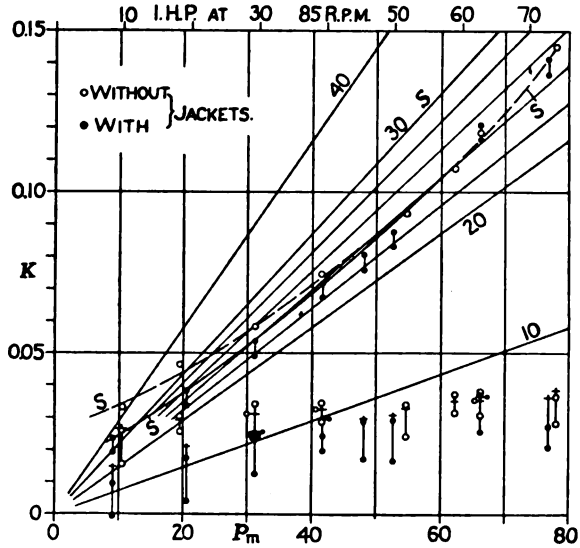

FIG. 95. — Tests without and with Jackets, 9 by 36 in. Corliss Engine, speed about 85 r.p.m., initial pressure 112 lb. abs., exhaust pressure 6 lb. abs. (these are pressures in the cylinder — the engine was run condensing, but with a poor vacuum). High-pressure section (alone) of the triple-expansion engine at Sibley College, Cornell University. R. C. Carpenter, Trans. A. S. M. E., Vol. 16, 1895, page 924. Range of cut-off, 0.01 of stroke at 66 lb. abs. to 0.40 at 104 lb. Clearance 0.076. For scheme of diagram, see Figs. 87, 88.

Fig. 96 represents tests made by Mr. Willans upon one of his central-valve engines, of small size. Those shown here are under the disadvantage, for present purposes, that change in cut-off is accompanied by change in one of the limiting pressures, either p_1 or p_2, sometimes in both. The first three groups in the first report, on noncondensing tests, are plotted in Fig. 96, where the ordinates are marked with the original designating numbers. According to this notation, the first number is the intended initial absolute pressure in the cylinder, the second the intended ratio of expansion. It appears that K_{mf} runs low in the simple engine, high in the compound and triple. Because of the high speed (400 r.p.m.), the missing quantity is relatively small, therefore the differences between K_{mf} and K_{mc} are of a low order of magnitude in comparison with the total steam measured. These are feed-water tests of a small engine, and the probability that the irregularities in the differences between the K_m's are due to small errors in the feed-water measurements makes it hardly worth while to attempt any deductions from these differences.

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Largely with the purpose of showing the kind of results got under service conditions, with large and well-maintained engines, the group of tests in Fig. 97 is here presented; as comparing performance with

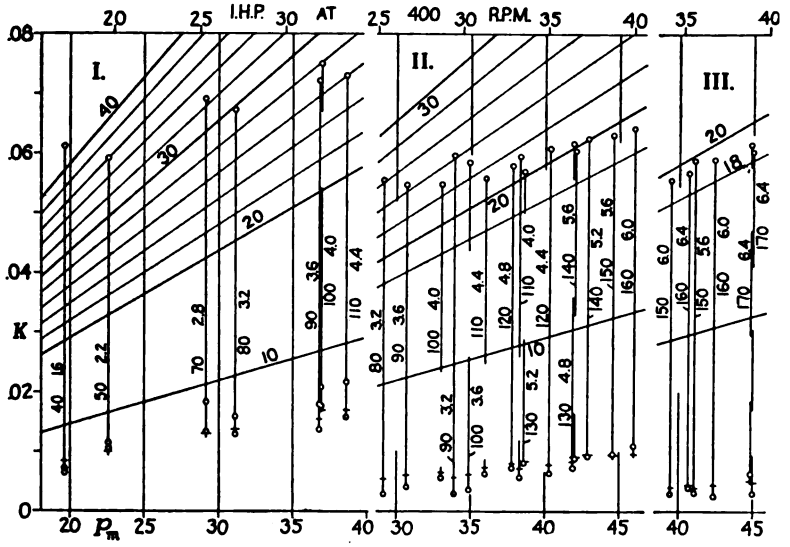

FIG. 96.— Tests of 7, 10, and 14 by 6 in. Willans Central-valve, Single-acting Engine. Group I, low-pressure cylinder, simple; group II, intermediate and low-pressure, compound; group III, whole engine, triple expansion. Clearances, high to low, 0.125, 0.105, 0.058. Engine run with late cut-offs and small compression. The basal m.e.p. is referred to the large piston in every case, and K is per cubic foot of displacement by that piston.

The tests here given are from the first report, Proc. Inst. C. E., Vol. 93, 1888, page 128. A second report, on tests with the engine run condensing, is published in Vol. 114, 1893, but data for an investigation of the missing steam quantity were not included in the matter printed.

and without jackets, this diagram might be placed in § 27 (*e*) or (*f*). Here we note that the lower values of the missing steam quantity agree well with the formula, Eq. (126); and that with the help of the jackets the engines are able to carry low loads, about one-fourth of their rating with but little increase in the steam per horse-power-hour. The relatively high values of K_{mc}, as also of steam consumed, indicate leakage from or past the high-pressure cylinder; Engine B is the worst in this respect, and A comes next.

(*g*) THE INFLUENCE OF CUT-OFF. — This appears to be very fairly represented, in its effect upon the ratio m, by the use of $\sqrt{e}$ as a divisor in Eq. (126). As regards the actual quantity K_{mc}, directly proportional to the missing steam at cut-off per revolution, the tests exhibited show that this changes very little with cut-off, keeping nearly constant over the range of ordinary working of the engine. It generally seems to

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fall off a little with very short admission; and at the other end of the range, with very late cut-off, a more rapid decrease occurs, at least according to Figs. 87, 88, and 91. The last point cannot be considered as fully established, since some of the evidence is contradictory.

As would naturally be expected, the reëvaporated steam increases steadily in amount with shorter cut-off and longer expansion. Figs. 95 and 97 show how this action is strengthened by the steam jacket.

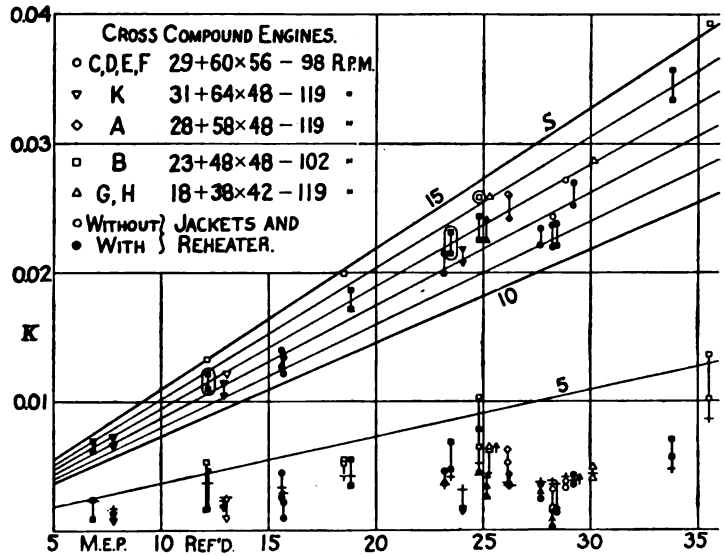

FIG. 97. — Tests of Large Vertical, Compound, Four-valve Generator Engines, made with and without jackets and reheater in action; tests under service conditions, but with load held steady in each case. Nominal gage pressure at throttle, engines A to F, 160 lb., engines G to K, 135 lb. Scheme of diagram, same as in Fig. 88. Report by L. S. Marks, Trans. A.S.M.E., Vol. 25, 1904, page 443.

(*h*) EFFECT OF THROTTLING. — Taking throttling to mean the cutting down of the initial pressure by a throttle valve in order to control the power of the engine, it is a subject which for detailed analysis, if that were worth while, would come later—this because it involves change in quality of entering steam and in range of pressure — but considered only as to general effect it appropriately follows the discussion of cut-off. Fig. 98 shows how initial condensation is diminished and finally abolished by throttling; deducting the clearance steam in test 35, the expansion curve would just about agree with the saturation line. As to whether throttling is economical, compare test 31 in Fig. 90 with test 35 here. There the m.e.p. is reduced to 11.7 lb. and the output per pound is 63.6 B.t.u.; here, with reduction to 13.2 lb., AU has been cut down to 46.0 B.t.u. The conclusion is obvious; but while strong

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throttling results in decided loss, through the throwing away of so much available work, there is every reason to believe that moderate throttling is advantageous in combination with early cut-off for very light loads.

Throttling as a means of control has its proper place in locomotives, and in other engines which must start frequently and under load. Even with a duplex quarter-crank engine, the cut-off must be late if the engine is to be able to start from any position. Under ordinary conditions, the locomotive has not enough "adhesion," or friction between wheels and rails, to resist the turning effect of full steam pressure upon the pistons throughout their stroke; consequently, some throttling is necessary, to prevent the wheels from letting go and slipping freely. In general, from a mechanical viewpoint, a nearly uniform driving pressure, such as appears in diagram 34, Fig. 98, is better at low speed than the kind of pressure variation that comes from early cut-off. As a method of governing stationary engines, throttling is justified only when simplicity and low first cost of the machine outweigh economy of fuel as determining considerations.

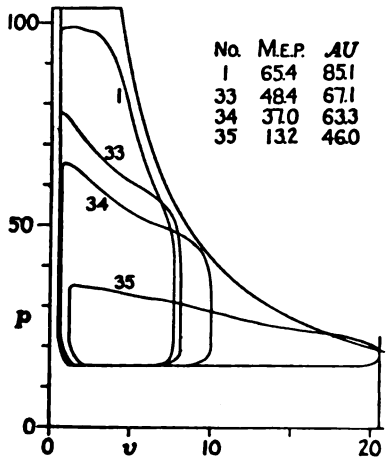

FIG. 98. — Unit Diagrams from Engine of Figs. 87 and 90, showing effect of throttling. Test 1 from Fig. 90; in tests 33, 34, and 35, cut-off is at 0.875 of the stroke.

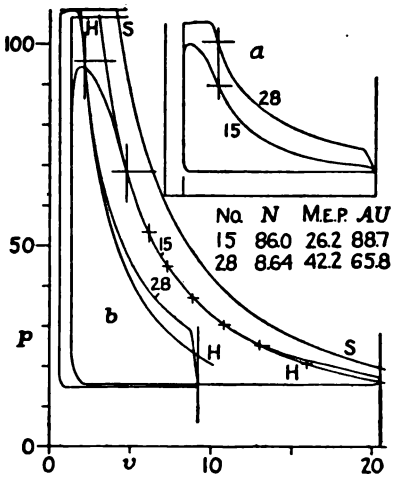

FIG. 99. — Diagrams from Engine of Figs. 87 and 90, showing effect of speed.

(i) EFFECT OF SPEED. — This is strikingly shown in Fig. 99, with speeds in the ratio of 10 to 1. The valve action is the same in both cases, but at the low speed much more steam is admitted and a fuller diagram and higher m.e.p. are obtained, as appears at *a*. The unit

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diagrams at *b* show, however, that the initial condensation is far greater at the low speed, and the effective output per pound of steam is much less. Hyperbolas are drawn along the expansion curves, originating at cut-off, and the difference in reëvaporation is thus clearly shown up.

(*j*) SPEED AND CYLINDER ACTION. — Figs. 100 and 101 are given to show how the missing quantity varies with speed, and to test the correctness of $\sqrt[3]{N}$ as a term in the condensation formula. The quantities plotted at *a* are K_{mc} and K_{mr}, with K_{mf} also in Fig. 101; the last is not marked off directly in Fig. 100, because of the crowding of the points. In Fig. 100 the steam pressure is about uniform, but there is

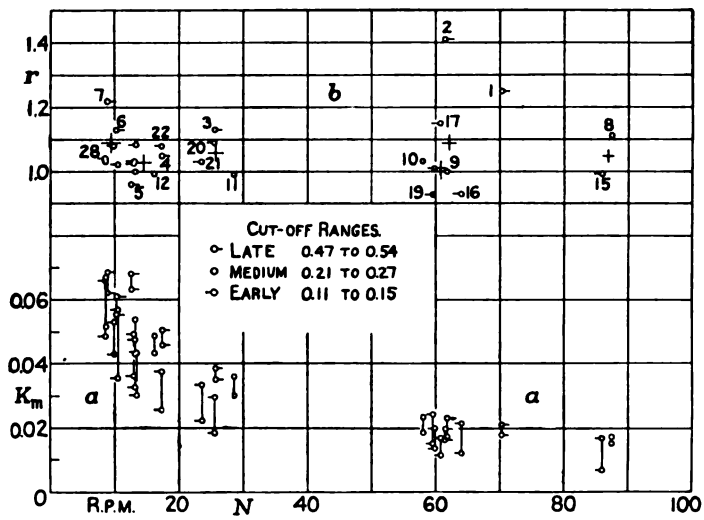

FIG. 100. — Variation of the Missing Steam Quantity with Speed. Same engine as in Figs. 86, 87, 90, and 99. At *a*, the missing quantity K_m; at *b* the ratio $r = m_f/m_c$, of value by Eq. (126) to observed value at cut-off.

a wide range of cut-off: note how the vertical lines connecting the two points, which show the amount of reëvaporation, increase in length with earlier cut-off, as well as with lower speed. In Fig. 101 the formula runs high, especially with the higher initial pressure, but follows very well the trend of K_{mc}.

To test the speed function in Eq. (126), the ratio $r = m_f/m_c$ is also plotted. In Fig. 100, individual values are given, and of course show a good deal of irregularity. By averaging in five groups and indicating the means by + marks, a better basis of judgment is established. For the group near 60 r.p.m., two mean values are shown, the higher got by including tests 1 and 2, in which *r* runs high for causes other than the

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speed — see Fig. 87. The mean points are so nearly on a horizontal line as to confirm strongly the use of $\sqrt[3]{N}$.

In Fig. 101, means of r for the five groups are plotted near the base line. Here it appears that N ought to have an exponent a little greater than $\frac{1}{3}$: with a larger value, so that the divisor would be somewhere

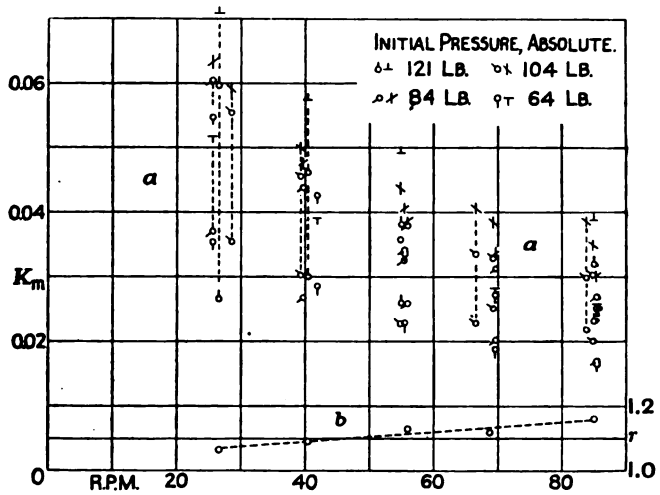

FIG. 101. — Speed Tests of 9 by 36 in. Corliss Engine, the same as in Fig. 95: tests at page 938 of reference there given. Engine run condensing, exhaust pressure 2.3 to 4.3 lb. abs. in cylinder; cut-off constant at 0.4 of stroke; initial pressure at four different values, as indicated on diagram.

between $\sqrt[3]{N}$ and $\sqrt{N}$, higher speed would exert a stronger influence, and keep m_f from increasing too rapidly.

(*k*) RANGE OF TEMPERATURE. — As implied in Art. (*c*), the amount of initial condensation is not in a simple proportional relation to the range of steam temperature within the cylinder. For presenting data along this line, and at the same time testing the sufficiency of the artificial temperature function T in meeting observed conditions, the scheme of Fig. 102 is effective. The ordinate is the ratio $r = m_f/m_c$, as used in the last two figures, and the base is absolute steam pressure. For any test or group of tests averaged together, the line 12 has its height fixed by r and is determined horizontally by the range of pressure within the cylinder, from initial p_1 to p_2 during exhaust. If r runs high, the formula is too strong and should be weakened, and vice versa.

From Mr. G. H. Barrus' book of *Engine Tests*, Fig. 102 represents a combination of all the results from engines of the Corliss class which were "tight" or "fairly tight" against leakage when inspected standing after the manner described in Art. (*p*). As would be expected in

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tests made under a variety of plant conditions, the points scatter pretty widely on a plot of individual values, r ranging from 0.78 to 1.35 for

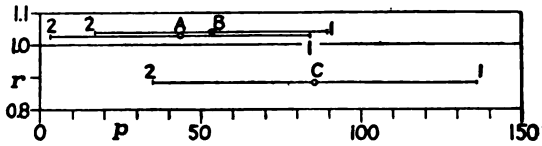

FIG. 102. — Ratio of Formula Value to Observed Value of the Missing Quantity, referred to the range of pressure within the cylinder. Tests of Corliss and equivalent engines, without jackets, in Barrus' *Engine Tests*.

A. Simple condensing engines, nine tests.
B. Simple noncondensing engines, ten tests.
C. Compound engine, high-pressure cylinder, nine tests.

the whole group. The use of mean values is proper with such data, in that it tends to balance errors and erratic variations and bring out any underlying trend. So far as simple engines are concerned, the formula averages up very well, running just a little high, as it does also with the engine of Figs. 86, 87, etc. For the compound engines, m_f at high-pressure cut-off does not work out big enough, in spite of the increase in range effected by substituting T from Table 10 for the actual temperature t — the amount of the change being shown in the following tabulation, made for the average pressures in Fig. 102:

	$t_1 - t_2 = t$			$T_1 - T_2 = T$		
Group A.....	316	144	172	315	184	131
Group B.....	320	220	100	321	214	107
Group C.....	351	260	91	361	246	115

A similar plot of mean results from six of the locomotives tested in the Pennsylvania Railroad Company's Testing Plant at the St. Louis Exposition of 1904 is given in Fig. 103. This involves the work-up of indicated steam consumption from the numerical data in the published book of reports: here engine No. 4 is omitted because it seems to show excessive leakage, No. 7 because superheated steam was used. The same general trend is evident as in Fig. 102, the shortening of the range of pressure and temperature exerting too much influence in the formula. Leakage is probably in large degree responsible for the very low value of r in cases 6 and 8. In getting this ratio, the factor m from Eq. (126) is applied to the total steam in the cylinder, including clearance steam, and the resulting missing quantity is divided by the actual $(K - K_{ic})$.

That data along this line are likely to be contradictory is well illustrated by Fig. 104. The series of points lettered A**B shows mean

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values of the ratio r from the four groups of tests in Fig. 101, where a simple condensing engine was run with the initial pressure at about 121, 104, 84, and 64 lb. abs.: each set of variable-speed tests at a particular pressure is averaged together. It appears that the formula gives too much weight to the longer range of pressure and temperature, not

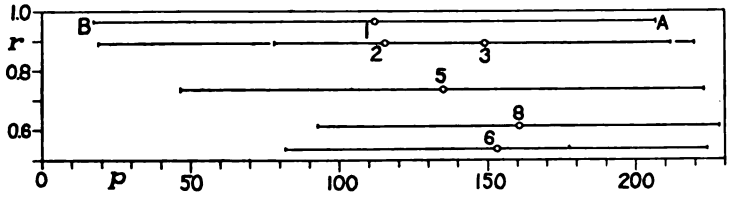

FIG. 103. — Ratio Diagram, Pennsylvania Railroad Locomotive Tests of 1904: see Figs. 105, 132, 133, 134. Engine sizes as follows:

1. Simple, two-cylinder, 22 by 28 in., freight service.
2. Simple, two-cylinder, 21 by 30 in., freight service.
3. Compound, two cylinder, 23 and 35 by 32 in., freight.
4. Compound, four-cyl. tandem, 19 and 32 by 32 in., freight.
5. Compound, four-cylinder "balanced," express service, 14.2 and 23.7 by 26.3 in.
6. Compound, four-cylinder "balanced," express service, 15 and 25 by 26 in.
7. Compound, four-cylinder "balanced," express service, 14.2 and 22.1 by 23.6 in.
8. Compound, four-cylinder "balanced," express service, 15.5 and 26 by 26 in.

enough to the shorter, which is in accord with the showing of Figs. 102 and 103. On the other hand, the three points S, C, and T represent results from the same engine, when run simple and condensing, and when serving as the first cylinder in a compound and in a triple-expansion: the first group is diagrammed in Fig. 95, but for present purposes only tests without the steam jacket are used, and those at very

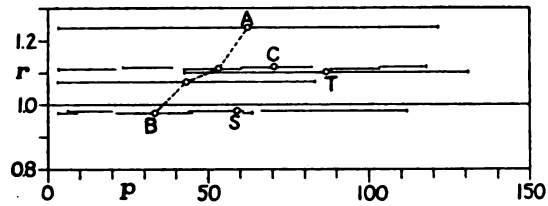

FIG. 104. — Ratio Diagram, 9 by 36 in. Corliss Engine at Sibley College, also in Figs. 95 and 101.

early cut-offs are omitted. These three group-averages have no very definite trend away from Eq. (126); and in this they agree with results from the 17 by 30 noncondensing engine of Fig. 86, etc. With the latter, tests were run at 60 and at 30 lb. gage pressure, in addition to the main body of tests at 90 lb. by gage; and for these Eq. (126) keeps m_f in very good alignment with m_c.

It is of interest to compare two of the engines just cited, in regard

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to the variation of m_c with initial pressure. That of Fig. 104, in the A**B series of tests, makes m_c increase as the steam pressure is lowered; in other words, the walls condense a larger fraction of the smaller weight of steam that comes in when the pressure and density are low even though the absolute weight condensed is less than at high pressure. In the noncondensing engine of Fig. 86, which agrees with Eq. (126), m_c keeps practically constant with changing initial pressure.

(*l*) INFLUENCE OF TEMPERATURE RANGE. — Reasoning from the information just presented, and anticipating some of that set forth in § 25, this matter may be put into the following fairly rational shape:

In the working of the engine between the pressure limits p_1 and p_2, a cyclical variation of steam temperature through the range $t = t_1 - t_2$ is set up within the cylinder; and from this results the cyclical absorption and rejection of heat by the metal surfaces.

If these surfaces followed the steam in temperature, either exactly or with a fair degree of proportionality, the amount of steam initially condensed would be proportional to the range t, or to some simple mathematical function of t, such as the square root. This hypothesis is not supported by the facts.

Experimental measurements of the temperature cycle of the cylinder walls — see § 25 — have shown that the temperature range of the surfaces is much smaller than that of the steam, and that the mean temperature of the metal is a good deal above a time-base mean of the steam temperature. This condition can be accounted for by the very reasonable assumption that there is a greater freedom of heat transfer from dense high-pressure steam to the metal than from the metal to the low-pressure steam of late expansion and of exhaust — see fuller statement in § 25 (*j*).

The state of affairs shown in Figs. 102 and 103 will be made rational by the hypothesis that, as the pressure limits p_1 and p_2 are changed, the range of wall temperature does not remain in constant ratio to the range of steam temperature, but varies at a less rapid rate. In other words, a moderate range of p and t, as from initial pressure to receiver pressure in the first cylinder of a compound engine, may set up a wall cycle of relatively high intensity: but an increase of steam range, as by dropping to atmospheric exhaust from this cylinder, will add very little to the amount of heat taken up and rejected by the cylinder walls.

The change from the actual range of t to that of the quantity T in Table 10 is a purely empirical attempt to meet existing conditions; and this function T has been chosen, after trial, in preference to the obvious scheme of damping the influence of t by adding a constant, or of using $(t + C)$ in the condensation formula. As between condensing and non-

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condensing simple engines, T seems to meet requirements quite successfully; for high-pressure cylinders the other expression, perhaps $(t + 100)$, might be better. It will not do to make the T-curve in Fig. 89 rise more steeply at high pressures, for that would give the temperature range entirely too strong an influence in an engine with a very long pressure range, such as the simple locomotive in Fig. 105. The practical expedient is to change the coefficient C, fixing different values for particular conditions: thus if Fig. 102 is truly typical, C should be made about 0.31 for compound engines of the class represented. Only by such adjustment can class peculiarities be taken into account; and the essential prerequisite is a good body of data from engines of the class under consideration.

(*m*) ENGINES WITH LARGE COMPRESSION. — This class is well represented by the locomotive, with the Pennsylvania Railroad tests of 1904, already instanced in Fig. 103, offering a large body of very complete data. Four sets out of the eight are plotted in Fig. 105, according to a scheme especially useful in the case of compound engines, for which that of Fig. 87 would become confusing. The diagram is based on the quality curves FF and GG in Fig. 82, but only at the "critical" cut-off and release points is the condition of the steam shown. The ordinate is absolute pressure, the abscissa the fraction of the working steam, or of the steam supplied to the cylinder, which is accounted for by the indicator. The clearance steam is not included in this ratio, so that m_t, indicated by cross-marks at cut-off, does not come directly from Eq. (126) with this class of engines, but must be worked out by the method described under Fig. 103: that is, the total steam is multiplied by m from the formula, then the resulting K_{mf} is divided by the admitted steam K. The experimental points belonging to a particular test are strung together by a broken line, but the segments of this line have no physical meaning.

For Fig. 105, the tests at the same speed are averaged together, so that in most cases a group of three to five tests is represented. Point A shows the pressure in the steam pipe and the practically unit quality of the steam coming to the engine. On the pressure line at cut-off B, a cross-mark shows the formula value, obtained as just described. On the line from high-pressure release C to low-pressure cut-off D, a cross-mark F indicates the lowest exhaust pressure in the high cylinder; for the simple engine No. 2, F is the exhaust pressure in the single cylinder. From D to E are shown the range of pressure and any change of quality during low-pressure expansion.

The actual size of the missing fraction m is more valuable information than the ratio borne to it by the m_t from an arbitrary formula, so

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that Fig. 105 is a much more useful type of diagram than the preceding figures. For purposes of comparison, one simple engine is included; and the most striking thing brought out is the small gain in quality at cut-off that is realized from compounding. Engine No. 3 shows an advantage of from 3 to 10 per cent over No. 2, but in Nos. 5 and 6 the missing quantity is fully as large as in the simple engine.

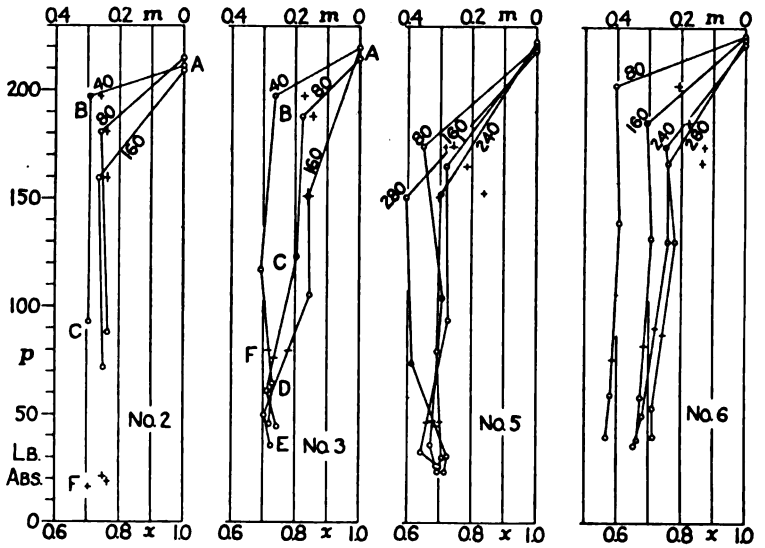

FIG. 105. — Quality Diagrams, for four of the locomotives listed under Fig. 103.

The basis of representation is partly responsible for the apparent failure of the missing steam to decrease as the range of pressure in the cylinder is made less by compounding: if the total steam were the base,

DATA FROM LOCOMOTIVES REPRESENTED IN FIGS. 103 AND 105.

Engine	1	2	3	5	6	8
Clearance	0.113	0.093	0.167	0.133	0.183	0.169
$\frac{K_c}{K}$ maximum	0.42	0.54	0.70	0.68	0.64	0.74
$\frac{K_c}{K}$ minimum	0.12	0.07	0.23	0.21	0.26	0.27

As regards the grouping in Fig. 105, it is to be noted that the ratio K_c/K will be higher as the speed increases, because the average cut-off is earlier at higher speed.

m would be smaller and would vary more. The clearance steam in the high-pressure cylinder has relative values of the kind shown in the tabulation just given, where K_c stands for clearance steam, calculated as if dry saturated at the beginning of compression, while K is the

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admitted working steam by feed-water measurement: the ratio is a good deal larger in the compounds. The uneconomical effect of excessive compression, more fully discussed in § 23, is strikingly apparent in these tests. Indicator diagrams from engines 2, 3, and 5 will be found in Figs. 132 to 134.

The marked discrepancy between formula and experiment is a strong indication of leakage, and a number of incidental inferences confirm this suggestion. Thus in engine No. 3, m increases from cut-off to release in the high-pressure cylinder (as against a normal decrease through re-ëvaporation), which would be accounted for by exhaust leakage, from the cylinder to the receiver; again, the quality in the low-pressure cylinder is almost independent of the speed, which is reasonable if relatively greater leakage at low speed balances increased cylinder-wall action. In No. 5 there is an anomalous increase of m with speed at 240 and 280 r.p.m., due perhaps to the development of some defect in the valves or in the piston packing as the tests proceeded; at 280 r.p.m. but one test was made, so that the apparent result lacks confirmation. With engine No. 6, the large size of the missing quantity seems to indicate copious leakage; but the fact that m varies so much with the speed in the low cylinder implies that this is valve leakage, — from live-steam supply right through to exhaust, — and the type of construction, with a single piston valve for the two cylinders together, rather favors this idea.

As regards the effect of speed, actual m_c and computed m_t keep in fairly constant ratio in each diagram, except for the marked discrepancy in No. 5, already alluded to, and a smaller divergence of the same sort in No. 2. The close similarity in the manner of variation of the two quantities in No. 6 weakens the hypothesis of excessive leakage, since that would exert a relatively greater effect at low speed, as appears in the high-pressure cylinder in No. 3.

(*n*) Influence of Size. — Direct experimental data upon this point, in the way of exactly similar tests upon exactly similar engines of different sizes, do not exist, and would be extremely difficult to secure: only by trial and inference can the correctness of the expression incorporated into Eq. (126) be judged. In apparently serving equally well for the small engines of Figs. 93 and 96 and for the large ones of Figs. 91, 94, and 97, the ratio s seems to meet the situation very fairly. Obviously, there must be some class variation, due to differences in arrangement of valves and ports and in the relative amounts of clearance surface. This can be found only by trial calculation, from the results of numerous tests, and will then be combined with the size factor, or else make itself evident in the "constant" C.

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The condition of the surfaces must have some influence upon the freedom of heat transfer, although probably not a great deal. Polishing the faces of piston and cylinder head has been tried as an expedient for diminishing cylinder action, but not with any striking result.

(*o*) VALUE OF THE CONDENSATION FORMULA. — The useful applications of this formulation of test results have been named in Art. (*d*): by giving a probable value of the missing quantity, it makes the engine an approximate steam meter, with data from indicator diagrams alone;

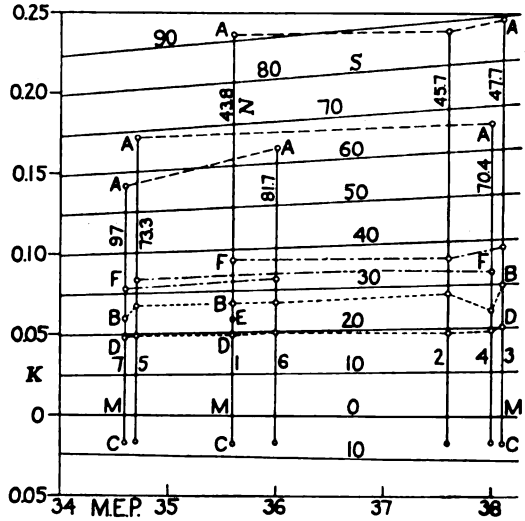

FIG. 106. — Callender and Nicholson Tests, Proc. Inst. C. E., 1897, Vol. 131. Engine 10.5 by 12 in., normal speed 250 r.p.m., clearance 0.10; shaft governor, valve as in Fig. 259. During experiments, governor was disconnected, eccentric clamped so as to give cut-off at about 0.2 of stroke, and speed was regulated (by a brake load, presumably) at the values marked along the ordinate lines, from 44 to 97 r.p.m.; engine was made single-acting by fastening to the valve an extension which kept the crank-end port always covered. Steam pressure 90 lb. by gage, exhaust atmospheric, into surface condenser.

and it serves as a check upon steam-consumption tests of working plants, referring especially to tests of the less precise type, such as are often the best that can be made under actual plant conditions.

In the great body of experiments upon which the formula is based, fairly represented by the examples which have been given, there must be considerable variation in quality of data. It is safe to say that in the best engine test that can be made there is, in the final relation of steam consumed to power developed, a possible error of 2 or 3 per cent; while under ordinary plant conditions, the uncertainty, even with reasonable care, may easily lie in the range from 5 to 10 per cent.

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Beside this possible error, there is reason to believe that some of the influences affecting the steam action are likely to fall into a state of rather unstable equilibrium, so that a slight preponderance or impetus in one direction will produce an erratic effect — witness the fact that it is found necessary to run the more complex engines for quite a time before beginning a test, in order to establish steady conditions and get consistent results.

Considering these uncertainties in the data, with the number and complexity of the influences which contribute to the net performance of the engine, too much must not be expected in the way of an accurate prediction. An estimate of actual steam would be made by dividing the indicated steam (at cut-off) by the fraction $(1 - m_t)$, having computed m_t through Eq. (126). Certainly, if the measured consumption differs from this estimate by more than 5 per cent on the side of deficiency or of 10 per cent on the side of excess, there is strong reason to question the accuracy of the test, or, in the case of excess, to feel assured that leakage is active. In engines with small compression, there will generally be better agreement than this; but with those having single slide valves and large compression, the net action seems to be more indeterminate and irregular, and is certainly more likely to be modified by leakage.

(*p*) Leakage. — The ordinary test for leakage consists in setting or blocking the engine in various positions and seeing whether steam escapes. With a slide valve, the wheel is turned until the eccentric radius stands about perpendicular to the stroke line, making the valve cover both ports, and the throttle valve is opened; separate admission valves can usually be released and moved into the closed position by hand. The indicator cocks may be used to see whether steam is getting into the cylinder, although it is better to take off the back head and get a clearer view, of one end at least: escape directly to the exhaust should also be looked for, with a single valve. Leakage past the piston and from the cylinder to exhaust is detected by blocking the wheel or crank and turning steam into one end of the cylinder. The dead center is the easiest position in which to hold the piston against steam force, and is sufficient for the test of the valve: but the piston should be tried at several points along the stroke line, if it can be properly held.

Of quantitative tests, with measurement of the amount of leakage, comparatively few are on record. Probably the most striking example is that diagrammed in Fig. 106. The main purpose of the experiments was the measurement of temperature, in metal walls and in steam — see § 25 (*c*) — but only the leakage determinations will be discussed at this point.

The diagram in Fig. 106 is plotted on the system of Fig. 87, with the

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steam K per cubic foot of piston displacement as ordinate and the mean effective pressure as base, so that all the quantities are shown also in pounds per horse-power-hour. The various amounts of steam laid out are as follows:

Measured from the base line MM, which is placed at the zero of the vertical scale, MA is the actual steam supplied, as condensed and weighed: MB is the steam actually kept in the cylinder for useful effect, deduced by subtracting the amount AB of leakage, as found by separate experiment. This leakage was chiefly past the long sides of the valve (not the ends, which cover the ports), directly from steam chest to exhaust: it was nearly constant per unit of time, hence the rate per unit of displacement varies rapidly with the speed of the engine, in inverse ratio. The valve was found to be practically tight when subjected to the standing test just described above, and the high rate of leakage was attributed to the breaking up of the oil film by motion; the steam was supposed to escape largely in the form of water, formed by condensation on the relatively cool edges of the exhaust port or chamber.

To return to the diagram, MC is the indicated clearance steam, c/s_c in Eq. (117), taken to be the actual weight, under the assumption that this steam is about dry at the beginning of compression; and CD is the gross indicated steam at cut-off, or e/s_e, so that MD represents the net indicated steam, K_{ic}: ME, shown for test 1 only, is the net indicated steam at release, or K_{ir}. From their temperature results the experimenters calculated the probable amount of steam actually condensed by the walls, which was of the same order of magnitude as the quantity DB.

To bring our missing-quantity formula into the discussion, the total indicated steam at cut-off, CD in Fig. 106, has been divided by $(1 - m_f)$ as got from Eq. (126); the result is laid off as CF, so that MF is the estimated steam consumption, to be compared with measured MB. If the indicated steam MD remained at about 20 lb. per h.p.h. when the engine was raised to the speed of 250 r.p.m., the consequent change in m_f would reduce the estimated consumption to about 28 lb., which would be very good performance for an engine of this size and class.

In these experiments, the valve leakage, already quoted as the principal component, was determined by plugging the steam ports with lead and running the engine with a motor, full steam being turned on at the throttle valve. Very nearly, the rate of escape, inferred to be mostly in the form of water, was found to vary directly as the perimeter of the valve and inversely as the width of overlap; and similar tests of other valves gave concordant results. With the ports open, the

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ends of the valve partly ceased to be available for leakage from live-steam to exhaust space; and uncertainty on this score raises a doubt as to the validity of the allowance which determines net steam MB from total condensed steam MA, as laid out in Fig. 106. In engines with separate steam and exhaust valves, this direct leakage to exhaust is impossible: and Eq. (126) is so largely based upon the performance of engines of the latter class that it probably covers but a small rate of leakage. In this connection, refer back to the remarks about leakage in Art. (*m*), under Fig. 105.

In general, leakage is something that can hardly be made predictable from determinate controlling conditions. The idea that it may assume considerable magnitude, even when the engine appears tight in a static inspection, has an important bearing upon the validity and usefulness of the method of thermal analysis described in § 24.

§ 23. Effects of Compression

(*a*) Action of the Clearance Steam. — If the steam in the cylinder was of the same average quality during expansion and compression, the curves of the two operations being essentially similar in form, the wasteful effect of clearance could be almost completely neutralized by compression. Turning back to the Rankine cycle, § 15 (*d*), we can readily see that a body of clearance steam might be so handled as not to affect the unit output or the efficiency of the cycle. This steam would be trapped in the cylinder at the end of exhaust and, further, would have just enough heat abstracted from it at the lower temperature t_2 to make adiabatic compression bring it to the state of the entering steam.* Then in expansion the clearance steam would perform all the work received in its compression, and with complete expansion its net work effect would be zero. Such a similarity of the two operations cannot be realized in the actual engine, so that the loss of available work which necessarily results from the presence of clearance can in but a small degree be diminished by compression.

As already briefly suggested in § 19 (*k*), the clearance steam (as a part of the whole body of steam in the cylinder) has a lower quality during expansion than during compression, hence does less work than it receives, causing a net loss. Using for illustrative purposes a steam diagram of actual form, the ideas set forth in § 19 (*g*) to (*k*) will now be worked out to a definite conclusion, which will show the best possible or imaginable result from the use of compression. The modification

* This would not be the same as the adiabatic compression of the whole charge in the Carnot cycle.

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by wall action will then be introduced, leading to the true relation, which is amply confirmed by experiment.

(*b*) THE FIRST ASSUMPTION. — In Fig. 107, with a constant admission and expansion line ABCD, the back-pressure line DEF is varied

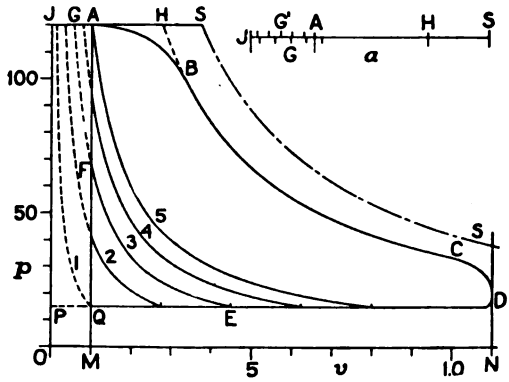

FIG. 107. — Steam Diagram with Variable Compression.

by changing the amount of compression, with equal increments, from zero to enough to raise the terminal point F up to the initial pressure at A: for convenience, curves BC and EF are of the form $pv = C$. The diagram is drawn to scale for one pound of total steam expanding, the full volume of which would run out to the saturation line SS; and at cut-off B the quality of the whole steam in the cylinder is 0.75.

At first we ignore SS and disregard initial condensation. After extending curves CB and EF up to the line JH, the length GH is taken to represent the volume (and weight) of the steam admitted from the boiler: division of each HG by the full one-pound volume JS gives the weight of this apparent or indicated working steam which corresponds to each of the five degrees of compression.

By a method described under Fig. 114 and in § 32 (*g*), an area scale of so many B.t.u. to the square inch can be calculated for the diagram in Fig. 107; then from planimeter measurements of the several ABCDEF areas, the cycle output AU (as in Fig. 90, etc.) is found for each case. Division of this AU by the weight found from GH gives the output per pound of steam admitted, expressed in heat units. In Fig. 108 the curve AB is laid out with this latter quantity as ordinate, the base QE being the same as QE in Fig. 107, so that the abscissa is the length along the stroke line, or the volume, through which compression takes place. The maximum of the curve, located graphically at M, shows the best degree of compression under the assumption which governs the determination.

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To cover the case of no clearance, area JABCDP is similarly compared with the steam of volume JH: the resulting unit output fixes the height of line CD. A vertical intercept between CD and AB shows the loss due to clearance. For this first assumption, if the expansion was complete, curve BC (Fig. 107) continuing without change of form until it met line PD, curve AB (Fig. 108) would have its maximum at B, and would there touch the line CD, showing complete neutralization of clearance waste.

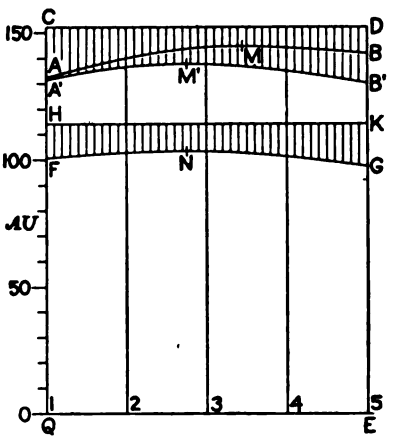

Fig. 108. — Curves of Performance, for ideal and actual conditions.

(c) Real Character of the Compression Effect. — Now go back to Fig. 107 and take into account the important fact that at cut-off — whether actual at B or effective at H — the steam in the cylinder has some fractional quality x_1, here 0.75. Assume that the clearance steam is dry saturated at E, so that its true weight can be calculated from the volume PE. Carry the several clearance-steam weights up to the line JS, laying off the corresponding full volumes in the lengths JG on the enlarged diagram at *a*. Also, multiply these JG volumes by x_1 and get JG′, which shows the clearance steam as of the same quality as the total steam at H. Note that these points are entirely independent of the form of the compression curves, which in its detail is an uncertain matter, determinable only by experiment, and unknowable in the extension FG dotted in upon the main figure. The length JG′ represents the effective service of the clearance steam, in filling space that would otherwise have to be filled with fresh steam from the boiler.

To get what will be a very close approximation to actual performance, we keep the AU values found from the areas ABCDEF, but use the new values of the steam weights which are represented at *a*. As a first step from the curve AB in Fig. 108, the true weights of the apparent working steam are found from the volumes G′H; and with these weights as divisors the curve A′B′ is obtained. Note the decided diminution in the amount of desirable compression, as determined by the maximum M′, also how much greater loss will be caused by excessive compression. The final curve FG is similar to A′B′, but is based on the real steam weight, volume SG in Fig. 107*a*, instead of the indicated steam: the same AU values are divided by the weights from SG and SJ, giving the ordinates of the curve FG, in

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Fig. 108, and the height of the reference line HK for the case of no clearance.

In effect, this is nothing but a comparison of indicated work with steam consumed. As Fig. 107 is laid out, two simplifying approximations are incorporated into the discussion: the first, however, or the use of similar curves, is wholly a matter of convenience, and the method is equally available when the compression curves differ from the expansion curve and among themselves. Of much greater importance is the use of a single, unchanged expansion curve BC as related to the unchangeable SS line for the pound of total steam, since this involves the assumption that the initial condensation (and subsequent quality variation) of the whole steam is independent of the relative amount of clearance steam. Some experimental information upon the last point is now in order.

(*d*) BEHAVIOR OF THE COMPRESSED STEAM. — This is strikingly shown in Fig. 109, by results from experiments in which the conditions

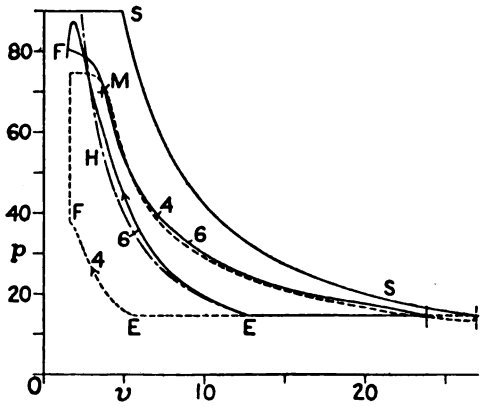

FIG. 109. — Unit Indicator Diagrams from Tests by Professor Dwelshauvers-Dery upon an 11.8 by 23.6 in. engine at the University of Liège. *Revue de Mécanique*, 1897, Vol. 1, page 939. Clearance 0.066, cut-off at 0.1 of stroke, compression through 0.1 and 0.4 of stroke: speed 61.7 r.p.m., saturated steam, atmospheric exhaust, no jackets.

were intentionally so arranged as to exaggerate the effects of compression. The diagrams are laid out for one pound of total steam expanding, that is, of clearance steam plus admitted steam, and SS is the saturation line for the pound of steam, as in Fig. 107. Redrawn from rather small printed diagrams, those in Fig. 109 are not of high accuracy in the minor details, but the general form is reproduced with essential correctness. Later experiments on the same engine — see § 25 (*i*) — have shown that the clearance steam was superheated at the begin-

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ning of compression, and throughout the greater part of that operation, which because of the consequent low rate of heat transfer was in large degree adiabatic. This is made apparent in test 6, by drawing a curve EH of constant steam weight — not greatly different from a curve of (moderate) uniform superheat — from the beginning of compression: the indicator curve rises decidedly above this reference curve until heat abstraction by the walls becomes active; while after condensation is established the compression curve droops so as to form a hook, showing a very rapid shrinkage of the product-measure pv.

Expansion and compression curves of the relative form shown in test 6, Fig. 109, are often obtained from engines of the single-valve, shaft-governor type — see § 39 (*g*) — when running with a very light load or with only their own friction load, this valve gear giving long compression with very early cut-off. The peculiarities here evident in such exaggerated form represent influences which are always present, but which under ordinary conditions of working are scarcely strong enough to make their effects apparent. In many engines, the compression curves are hardly distinguishable from the equilateral hyperbola; and frequently, with large compression, there is merely a small decrease in the product pv toward the upper end of the curve, as a slight approach to the droop or hook here shown. It is to be noted that only when taken with a reducing motion of assured accuracy may the compression curve of a diagram be subjected to really close analysis.

(*e*) Influence of Compression upon Expansion. — As bearing upon the question implied at the end of Art. (*c*), the two diagrams in Fig. 109 suggest the important practical conclusion that the condensation factor m for the total steam is essentially independent of the amount of compression — this for the case of constant cut-off in the cylinder, with compression varied. Incidentally, the mark at M on Figs. 109 and 111 shows the application of the factor m_f, as computed from Eq. (126), to the total steam at cut-off. In further illustration, Fig. 110 gives the results of the six experiments published, laid out on the compression base that is used in Fig. 108. The ordinate is steam per cubic foot of piston displacement, and the quantities plotted are as follows:

AB is the total steam, made up of clearance steam AD and admitted steam DB: the weight of the former was computed as for dry saturation at the beginning of compression. The effect of speed is clearly shown in the difference between the two BB lines.

AC is the gross indicated steam at cut-off: its (small) variation from constancy shows that the cut-off was not kept quite uniform in effective value.

CB is the total initial condensation, with the comparatively small

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variation just noted: the ratio $m = \overleftarrow{CB}/AB$ is more nearly constant than CB itself.

AE is the indicated clearance steam at the end of compression, or at the point F in Fig. 109. The difference DE shows the condensation of the clearance steam by the cylinder walls before admission begins: it is really a part of the total condensation BC, although not so placed on the diagram.

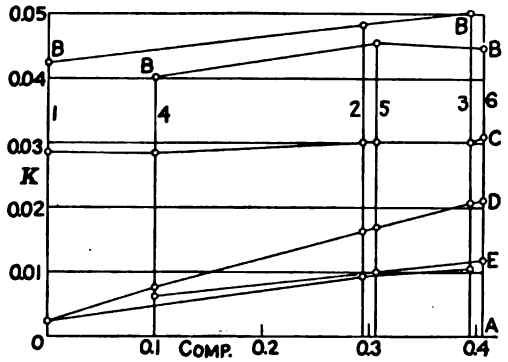

FIG. 110. — Steam-quantity Diagram of the Dwelshauvers-Dery compression tests. In tests 1, 2, 3, r.p.m. = 45, in 4, 5, 6, r.p.m. = 61.

It would naturally be expected that with earlier compression the amount of steam condensed by the cylinder walls will increase, because the time during which the steam is above the temperature of the walls is made longer. This expectation is met in Fig. 110, but the increase is relatively small, not big enough to interfere with the usefulness of the assumption of constancy in approximate discussions. This assumption underlies the application of the factor m from Eq. (126) to the total steam in large-compression engines, as described in the definition in § 22 (*b*) and exemplified in Figs. 103 and 105. In § 24 (*c*) it is shown that if the missing quantity at cut-off remains constant, there is implied a considerable increase, with compression, of the heat absorbed by the cylinder walls.

(*f*) CYCLE OF THE CLEARANCE STEAM. — Knowing the ratio of clearance steam to total steam, by weight, it is a simple matter to lay out the expansion curve of the quantity of steam that was compressed. For the two tests in Figs. 109 and 111, the value of this fraction, or of AD/AB in Fig. 110, is, respectively, 0.189 and 0.471. In Fig. 111, any volume ac of the total steam is multiplied by the factor just given in order to get the volume ae of the clearance steam.

It now appears that the clearance steam follows a complete cycle of its own: and since during compression the steam receives more work

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than it gives back during expansion, the cycle has a negative output, represented by the shaded area in each case of Fig. 111. With incomplete compression, test 4, the manner in which the new steam continues to compress the old steam, above F, is indeterminate; but since this action involves the kinetic loss explained in § 19 (*i*) and (*j*), it is proper to make the line FA serve as one of the boundaries of the lost area.

Of the shrinkage in effective output due to the harmful influence of the cylinder walls, a portion bears upon the clearance steam, as shown by these cycle diagrams, and only the remainder is carried directly by

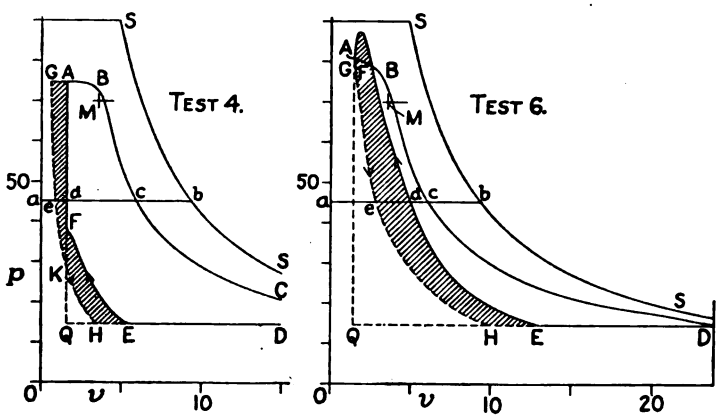

FIG. 111. — Cycle of the Clearance Steam, same diagrams as in Fig. 109.

the new, working steam; but ultimately all the loss comes upon the latter, whence results the fall in efficiency which occurs when compression is increased beyond a moderate amount under a certain cut-off. In slightly different statement, the total missing quantity with diagrams like Figs. 107 or 109 is in a nearly constant ratio to the total steam; hence its ratio to the working steam will increase rapidly as the latter is made smaller through excessive compression.

This is closely analogous to what takes place when cut-off is made earlier in a small-compression engine. As shown in § 22 (*d*) and (*e*), the steam condensed per revolution varies but little with cut-off, therefore as the latter is made earlier the relative magnitude of the missing quantity increases. It has been suggested that an engine with large clearance might be governed, in part at least, by varying the compression instead of the cut-off. Which scheme would cause the greater loss of efficiency, in the reduction of the mean effective pressure to a desired small value, is a question that could be answered only by experiment, because secondary actions of unknown amount would vitiate

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conclusions drawn from approximate deductions by methods like that of Fig. 107.

In terms of Fig. 111, test 4, the gain from a moderate compression continues so long as area AKG decreases more rapidly than EFKH increases.

(*g*) TESTS WITH VARIABLE COMPRESSION. — In Fig. 112 is plotted a set of tests made to determine the effect of compression upon economy. Here and in Fig. 113, the ordinate is steam consumption in pounds per horse-power-hour, in effect the reciprocal of the ordinate in Fig. 108, so that a minimum now shows best performance. In each series of

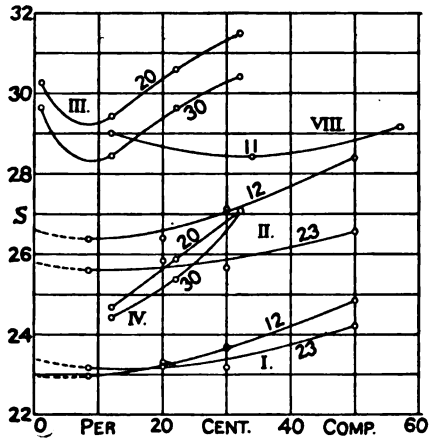

FIG. 112. — Tests on 7.1 by 17.7 in. Corliss Engine, at Dresden, A. H. Klemperer, *Zeitschrift des Vereines deutscher Ingenieure*, 1905 I, page 797.
Clearance 0.045, increased to 0.152 in series VIII; speed 101 to 102 r.p.m.
Series I, II, VIII, steam at 100 lb. abs., condensing.
Series III, IV, steam at 114 lb. abs., noncondensing.
Series I, jackets on; others, no jackets.
Series IV, superheat of 90 deg. fahr.; others, saturated steam with about 0.01 of moisture.

Fig. 112, the cut-off is held constant at the percentage of the stroke indicated by the number on the curve, and compression is varied. Between series I and II there is the difference in consumption due to the steam jackets (on heads and barrel), but the curves are very similar; in both cases, more compression is allowable with the later cut-off. Series III is the most complete, in that it contains tests with but 0.01 of compression, and shows an unmistakable minimum of consumption at about 0.09, or at twice the clearance volume. Series IV agrees with III in form, and differs by what might be expected as to the gain from superheating in so small an engine. Curve VIII shows tests made after the clearance had been increased to more than three times the

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normal value; now the best performance comes with more than 0.3 of compression.

The tests in Fig. 113, directly plotted at I, were made in series with constant compression and variable cut-off; here the numbers on the curves show the percentages of the stroke through which the steam is compressed. On the same base with I, lines relating m.e.p. to cut-off are plotted at II. From these two groups of curves are derived those at III, which relate consumption to compression when m.e.p. and i.h.p. are kept at a constant value. As an example, the horizontal line for m.e.p. = 30, in group II, cuts the m.e.p curve for 0.11 com-

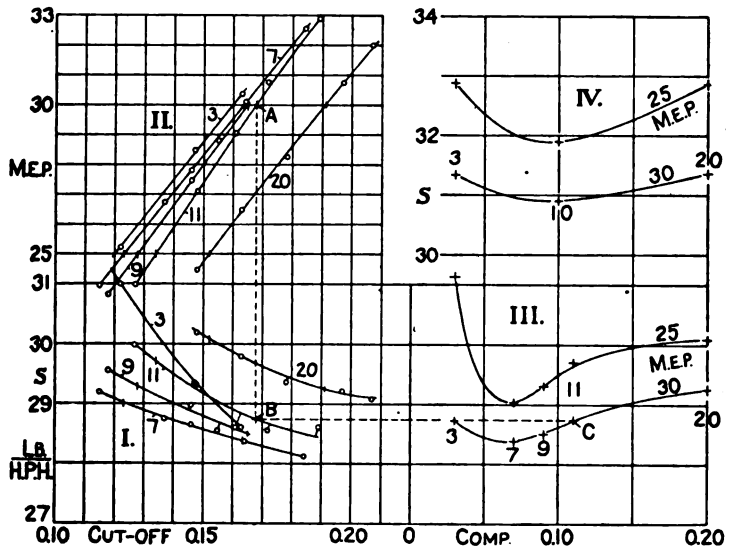

FIG. 113. — Tests on a 9.8 by 19.7 in. Corliss Engine at the University of Ghent. Professor J. Boulvin, *Revue de Mécanique*, 1907, Vol. 20, page 109.
Clearance 0.038, speed 101 to 103, steam pressure 93 lb. abs., noncondensing; m.e.p. 24 to 33 lb., i.h.p. 18 to 25.
Tests at I, II, and III, with jackets; curves at IV derived from tests without jackets.

pression at A, where the cut-off is about 0.168. Projecting A down to B, on the curve for the same degree of compression, we get the corresponding steam rate *S*. The latter is carried over to III, where the ordinate is located on a base of compression percentage. These curves show how consumption varies with compression, but do not indicate coincident cut-offs, leaving these to be read off from I or II. The curves at IV are similar to group III, but represent another lot of tests, made without steam in the jackets.

The primary curve 3 in group I is of such different form from the

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others as to appear somewhat doubtful, both in itself and in the very high points which it determines at the left-hand ends of the curves in group III. Of greater interest is the shape of the latter curves to the right of their minima: the way in which they first rise rapidly, then more slowly, needs explanation. This can be given in terms of the statement under Fig. 112, that more compression is economically allowable with later cut-off, which holds true until the incompleteness of expansion becomes considerable. With more compression there must be later cut-off for the same m.e.p.; then the condensation factor m is smaller, and the clearance steam is relatively less in amount. For a given compression, therefore, the negative work as in Fig. 111 is absolutely less and is deducted from the output of a larger weight of working steam.

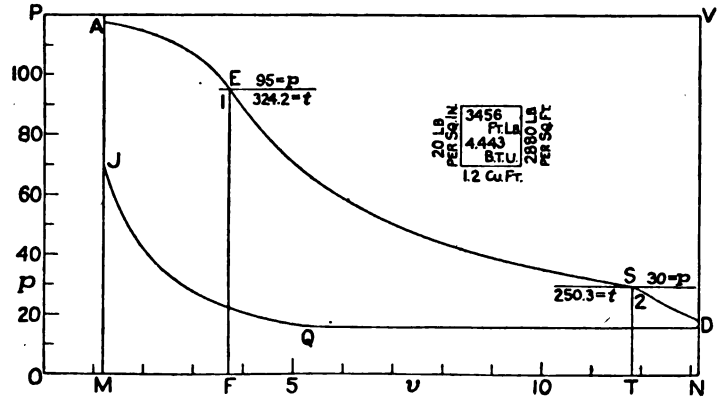

FIG. 114. — Data for Hirn's Analysis.

(*h*) CONCLUSIONS. — Merely for emphasis, the following conclusions, deduced in the course of this discussion, may be grouped together:

Clearance always causes a loss of effect, which can in but a small degree be diminished by compression.

Under any conditions, the introduction of some compression will be advantageous; but the best amount may be small, and is less as the conditions of working cause the initial-condensation fraction m to be greater.

In the earlier ranges, as the cut-off is shorter the desirable compression is less. The converse, that compression may increase with cut-off, holds until incompleteness of expansion begins to exert an opposing influence; so that there will be some cut-off for which the economically allowable compression will have a maximum value.

Taking Fig. 107 to be fairly typical in its proportions, we see that

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enough compression to meet mechanical requirements and promote smooth running — refer to § 34 (c) — even if it be in excess of the best amount from thermodynamical considerations, may be introduced with entailing appreciable or serious loss.

§ 24. Analysis for Thermal Effect

(a) Heat Added, Energy Change, and Work Done. — Take the fundamental statement of relation for a thermodynamic operation, Eq. (14),

$$Q = I + AU,$$

and modify its form by letting I_1 stand for internal energy at the beginning and I_2 for internal energy at the end of an operation, so that the energy-change I is equal to $(I_2 - I_1)$. Also, repeat the definition of AU as the heat value of the external work done in the passage from state 1 to state 2, and of Q as the heat imparted from without. Then the equation

$$Q = I_2 - I_1 + AU \quad . \quad . \quad . \quad . \quad . \quad . \quad (130)$$

furnishes a ready means of determining the amount of heat imparted to the steam in, say, the operation of expansion within the engine. An example will best illustrate principle and method.

(b) Calculation from the Indicator Diagram. — In Fig. 114 the volume scale is such that the diagram represents the performance of one pound of total steam. The clearance steam, taken to be dry at the beginning of compression at Q, is 0.206 of the total steam: then the diagram is for 0.794 lb. of working steam, and the clearance steam is 0.206 ÷ 0.794 or 0.26 of the working steam per cycle. At E the quality is $x_1 = 0.80$, at S it is $x_2 = 0.857$. The two values of internal energy are first computed with data from Table II by Eq. (79), $I = K - ml$, as follows:

$$I_1 = 1103.7 - 0.20 \times 809.6 = 1103.7 - 161.9 = 941.8;$$
$$I_2 = 1087.1 - 0.143 \times 868.4 = 1087.1 - 124.2 = 962.9:$$

then $(I_2 - I_1) = 21.1$ B.t.u.

By planimeter measurement, the work area ESTF is found to be equivalent to 75.3 B.t.u.: the dimensioned square, above the indicator diagram, represents one square inch on the original drawing, and shows how the area scale is derived from the linear units. Substituting in Eq. (130), we have

$$Q = 21.1 + 75.3 = 96.4 \text{ B.t.u.}$$

This is per pound of total steam in the cylinder; per pound of working steam it would be $96.4 \times 1.26 = 121.4$ B.t.u.

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In this assumed diagram, smoother in outline than if actually taken by the indicator, the expansion is made to follow exactly the curve $pv = C$. It is of interest to note the decided increase of internal energy that takes place; the heat returned by the cylinder walls is not only sufficient in amount to equal the external work done, but there is a surplus left over to be added to the thermal energy of the steam.

(*c*) HIRN'S ANALYSIS. — The relation stated in Eq. (130), applied in the manner just illustrated, is the working principle of what is called Hirn's analysis for the determination of the heat interchanges between the steam and the cylinder walls. In brief outline, the scheme is as follows:

The calculation is started at Q, the beginning of compression, usually with the assumption that the clearance steam is there dry-saturated, which makes its weight a known quantity — see § 25 (*i*) for some special information concerning the state of the steam at this point.

With steam weight, pressure, and volume known at Q and at J, the energies I_Q and I_J are easily found. To the initial energy I_Q add the piston work AU_{QJ} under curve QJ (measured down to the base ON and reduced to heat units); then if I_J is less than this sum, the difference has gone from the steam into the walls, if it is greater the difference has come into the steam.

At no point from J up to A and over to E is it possible to know just how much steam is present in the cylinder; but as soon as cut-off has made the steam weight definite, the calculation can be resumed. To the energy I_J of the clearance steam at J add the total heat H_1 which the new steam had when it entered the cylinder. From this sum subtract the energy I_E of the whole steam at E plus the piston work AEFM or AU_{AE}; the difference is the heat given to the walls from J to E.

In algebraic expression, the heat interchanges just described are,

$$Q_{QJ} = I_Q + AU_{QJ} - I_J, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (131)$$

$$Q_{JE} = I_J + H_1 - I_E - AU_{AE}; \quad . \quad . \quad . \quad . \quad . \quad (132)$$

and the sum of the two, or the total heat yielded by the steam during compression and admission, is

$$Q_{QAE} = H_1 + I_Q + AU_{QJ} - I_E - AU_{AE}. \quad . \quad . \quad . \quad (133)$$

To continue the numerical illustration in the last article, the data represented by Fig. 114 may be worked out as follows:

At Q, with $p = 15.7$ lb. abs., $K = 1077.6$, $x = 1.00$, $w = 0.206$ lb.,

$$I_Q = 0.206 \times 1077.6 = 222.0 \text{ B.t.u.}$$

The area under QJ is 5.11 sq. in., so that

$$AU_{QJ} = 22.7 \text{ B.t.u.}$$

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The entering steam is at 120 lb. abs., for which pressure the total heat is 1189.8; with 0.794 lb. of steam coming in, assumed to be dry-saturated,

$$H_1 = 0.794 \times 1189.8 = 944.7 \text{ B.t.u.}$$

The energy of the steam at E has already been calculated, in Art. (*b*), as

$$I_E = 941.8 \text{ B.t.u.}$$

By measurement, area AEFM = 11.56 sq. in., or

$$AU_{AE} = 51.3 \text{ B.t.u.}$$

Now substitute in Eq. (133) and get,

$$Q = 944.7 + 222.0 + 22.7 - 941.8 - 51.3$$
$$= 1189.4 - 993.1 = 196.3 \text{ B.t.u.}$$

At J, $p = 70$, $x = 0.915$, $I_J = 212.0$; then the heat given up during compression comes out as

$$Q_{QJ} = 222.0 + 22.7 - 212.0 = 32.7 \text{ B.t.u.},$$

leaving $196.3 - 32.7 = 163.6$ B.t.u. as the heat surrendered during admission.

An approximation which naturally offers itself is to take the latent heat of the steam condensed at cut-off (or of the "missing steam") as the heat absorbed by the cylinder walls. Only a very small error is introduced by carrying $m_1 = 0.2$ from E up to the initial pressure on line PV, where $r_1 = 877.6$; then $m_1 r_1 = 175.6$ B.t.u., as against 196.3 above. Obviously, as compression and throttling during admission are less, these two results tend to come into closer agreement; on the other hand, if the missing quantity keeps constant while compression increases — compare § 23 (*e*) — the heat absorbed by the walls will also increase, because the equivalent of the work of compression adds itself to the latent heat of the steam condensed.

Note that in this discussion the symbols I and AU are not restricted to one pound of steam, but are used in relation to any weight which may be present.

(*d*) Completion of the Cycle. — The calculation of the heat returned during expansion, according to the equation

$$Q_{ES} = I_S + AU_{ES} - I_E, \quad . \quad . \quad . \quad . \quad . \quad . \quad (134)$$

has been illustrated in Art. (*b*). Subtracting the 96.4 B.t.u. there found from the 196.3 which the walls have taken up, we have 99.9 B.t.u. of this heat yet left in the metal, barring loss by conduction and radiation. This is, of course, less than the $0.143 \times 944.5 = 135.1$ B.t.u. of latent heat of the moisture in the steam at S: necessarily, there must be some net condensation, because of the performance of the work of

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expansion. Complete reëvaporation is not needed for the assumption of dry steam at Q, since a considerable part of the water remaining at S will be thrown off the metal surfaces and swept out, in the rapid pressure drop and outrush of steam from S down to the exhaust pressure.

To complete the determination and get a "heat balance" for the cycle, the condenser must be made to serve as a calorimeter. From the weight of cooling water per pound of steam condensed, with its initial and final temperatures, the heat content of the exhaust steam is calculable, being finally expressed, like all the other quantities, as if measured above 32 deg. The energy I_S, plus the work AU_{SQ} done by the piston upon the exhaust steam, plus the heat further received from the walls, minus the energy I_Q, should just equal the heat content measured by the condenser. The heat from the walls is the "unknown" component, and with absolutely exact experimentation its true value would be made known: whereupon, any deficiency from the computed residuum could with confidence be charged to radiation. Actually, the best accuracy attainable will lead to but a very rough measure (proportionally) of these relatively small quantities.

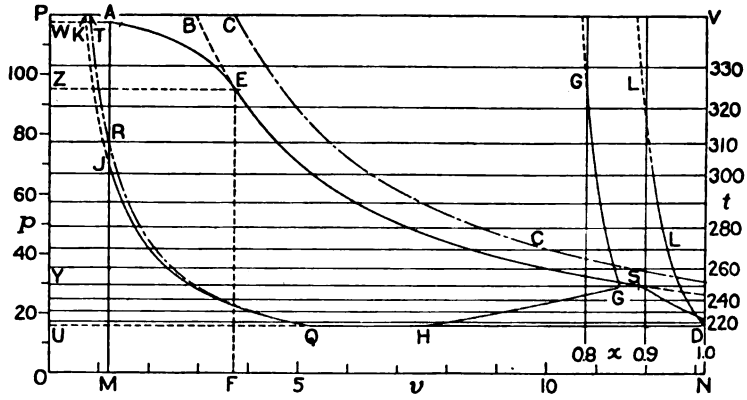

FIG. 115. — Data for the Temperature-entropy Diagram.

Granting full accuracy of indicator diagrams and of steam measurement, but having only these data, there are two points of uncertainty in this analysis; one is the real quality of the clearance steam, the other, the amount of leakage occurring. The determination of heat rejected, by means of the condenser, helps to settle the latter question, except as complicated by radiation loss.

(*e*) THE TEMPERATURE–ENTROPY DIAGRAM. — In preparation for its transfer to the temperature-entropy plane, certain preliminary work must be done upon the indicator diagram, as is illustrated in

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Fig. 115. At the right-hand end of that figure, a scale of corresponding steam temperatures is laid off, and isothermal lines are drawn across the diagram at equally-spaced temperatures. Two curves of constant steam weight are laid out, CC for one pound of total steam, QRT for 0.206 lb. of clearance steam — the diagram being the same as in Fig. 114. For the expansion and compression curves, equilateral hyperbolas are assumed; and this definite form having been chosen, the compression curve can be extended up to the line of initial pressure at K. Two curves of quality x are plotted at the right, GGH for expansion and release, LL for compression. A value of x on curve GG is got by dividing the abscissa of curve ES by that of curve CC; similarly, points on LL show the ratio of volumes between compression curve QJK and constant-weight curve QRT. For GH, an abscissa of curve SD is divided by the value of the steam volume s from Table II; calling this a "quality curve" implies the idea of the steam being condensed within the cylinder, roughly at constant volume; which imaginary operation is, in effect, substituted for that of actual outflow, when the diagram is changed over into the temperature-entropy system of representation.

The new diagram is laid out in Fig. 116. Having the x curves in Fig. 115, it is a simple matter to divide in the ratio x the entropy N_r or b, included between the curves WU and CC: measurement and slide-rule calculation is the preferable method. Diagram WESDU in Fig. 116 is for the total steam, representing the outline with the same lettering in Fig. 115. There is no way of transferring the closed indicator diagram AESDQR to the temperature-entropy plane without some distortion of the entropy abscissas. Even the curves WAE and SDU, easily enough laid out from volume ratios, do not by any means represent the actual processes of evaporation + inflow + cylinder-condensation and of outflow + condensation-of-exhaust: they merely stand for imaginary thermal actions which would have the same pressure-volume effects as are shown by the corresponding curves on the indicator diagram. Only when the performance of a confined body of steam is in question, or when the engine valve is closed during expansion and compression, can the temperature-entropy diagram be taken to mean exactly what it appears to say.

(*f*) Thermal Quantities for the Total Steam. — Before going on to see how the performance of the clearance steam may be represented, we shall consider the determination of heat quantities belonging to the total steam, as derivable from the profile PAES in Fig. 116.

If there were no clearance and compression, all of this steam coming in from the boiler and having at entrance the quality corresponding to the point C, the heat given up to the walls would be the difference

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between the area under UPC and that under UWAE — these areas being bounded by vertical ordinate lines from U, C, and E, and extending down to the line of absolute zero of temperature. The effect of compression, as set forth in Art. (*c*), cannot here be shown in any simple or effective fashion, so that resort to Hirn's method is necessary if the heat transferred to the cylinder walls is to be exactly measured.

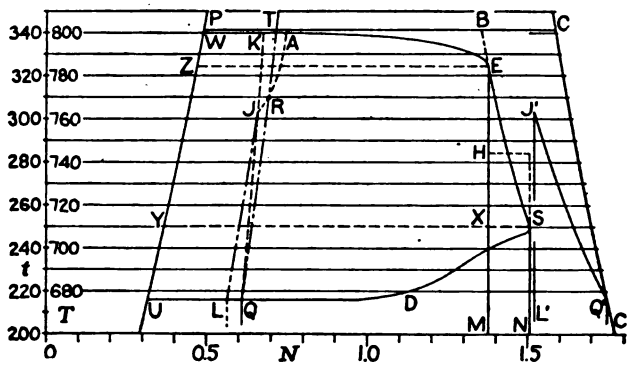

FIG. 116. — Temperature-entropy Diagram from Fig. 115.

As to the heat restored during expansion, the obvious thing is to get the value of the area under ES. By planimeter measurement of area ESX, the top of an equivalent rectangle is located at H, at 744 deg. absolute. The entropy width MN, measuring up as scant 0.129, is found by computation from tabular data, with $x_1 = 0.80$ and $x_2 = 0.857$, to be 0.1293. Then the heat imparted is $Q = 0.1293 \times 744 = 96.2$ B.t.u., as against 96.4 B.t.u. by the parallel computation in Art. (*b*). The discrepancy is well within the limits of error of the methods used.

Another scheme is based upon an extension of the idea of the Rankine cycle. Consider the diagram ZEXY in Fig. 116; in the cyclical lowering of pressure from ZE to XY, the external work represented by the area ZEXY is equal to the difference between total heat h_1 at E and total heat h_2 at X, as is brought out in § 15 (*d*) and § 16 (*b*). With some other expansion curve than the adiabatic, as ES, and with heat Q coming in while the steam is passing from E to S, the heat value of the work ZESY will be

$$AU_{ZS} = h_1 + Q - h_2;$$

whence

$$Q = AU_{ZS} + (h_2 - h_1). \quad . \quad . \quad . \quad . \quad . \quad . \quad (135)$$

In the particular case of Fig. 115 — that is, with an expansion curve of the form $pv = C$ — the difference $(h_2 - h_1)$ will be the same

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as $(I_2 - I_1)$ in Art. (*b*); also, the area between the curve ES and either axis will be the same. Other proportions would better have illustrated the general proposition that the area beneath an expansion curve plus the change of internal energy is equal to the area from the curve over to the vertical pressure axis plus the change of total heat.

(*g*) BEHAVIOR OF THE CLEARANCE STEAM. — The only way to get an exact representation of thermal action during compression, with entropy measured on the scale at the bottom of Fig. 116, is to draw the curve Q′J′ for one pound of clearance steam, using the x curve LL in Fig. 115. Then the area under Q′J′ is the heat given up by one pound of steam in being compressed along the curve $pv = C$, from initial dryness at Q — and similarly with any other actual form of curve.

It is a simple matter to lay out on the main diagram a curve which will represent the operations of compression and clearance filling, at least so far as to divide off an area corresponding with the indicator diagram: but this will not be a true temperature-entropy line, nor will the area beneath it, between vertical ordinates, be equal to heat transferred. Having 0.206 lb. of clearance steam in the example under consideration, draw a constant steam-weight line TQ on Fig. 116, dividing all the distances between WU and CC in this ratio. Then locate QJ with reference to QT and UW just as Q′J′ is located between CC and UW. In the diagram, the whole curve QJK is transferred from Fig. 115, likewise the line JA; the idea under all these constructions being to make volume ratios and entropy ratios equal. Of course, the curve JA, referred to the line QRT in Fig. 116, has no semblance of physical meaning; but the area AESDQJA is equivalent to the indicator diagram with the same lettering in Fig. 115.

It is possible to refer to the line QJ, Fig. 116, an area which shall represent the heat given up in compression. By the same method of proportions as above, transfer the constant-entropy line J′L′ to the space between UW and QT, where it appears as JL: then the area QJL plus the area under QL (the latter between verticals) shows the heat which must be abstracted from 0.206 lb. of steam in order that it may follow the curve QJ in being compressed through the range of pressure indicated on Fig. 115.

(*h*) UTILITY OF THE TEMPERATURE-ENTROPY DIAGRAM. — Several schemes for taking into account the performance of the clearance steam in the layout of this diagram have been proposed by different writers, but that represented by the outline AESDQJA in Fig. 116 is the one commonly used, and seems to be the best. Objections to it are found in the lack of correlation between the inner curve JQA and the funda-

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mental meaning of the coördinates of the diagram, and in the fact that with multiple-expansion engines the change in weight of clearance steam, from cylinder to cylinder, calls for all kinds of unsatisfactory expedients. Considering that the expansion curve is the only part of the profile with a real physical meaning, the diagram scarcely justifies itself as a means of exact thermal analysis and representation: certainly the method of Hirn is far clearer and more serviceable for the determination of the values of the secondary heat quantities. But if the point of view be somewhat shifted and a further slight sacrifice of strict construction be allowed, the temperature-entropy diagram can be made a very simple and useful device for comparing actual with ideal performance, or the indicator diagram with the Rankine cycle. The method now to be proposed bears some analogy to the elimination of the clearance steam from the pressure-volume diagram, as in Figs. 75 and 81, and the resulting diagram is for the working steam alone. In Figs. 117 and 118, only the full lines belong to the final scheme; the dotted curves merely illustrate certain secondary matters.

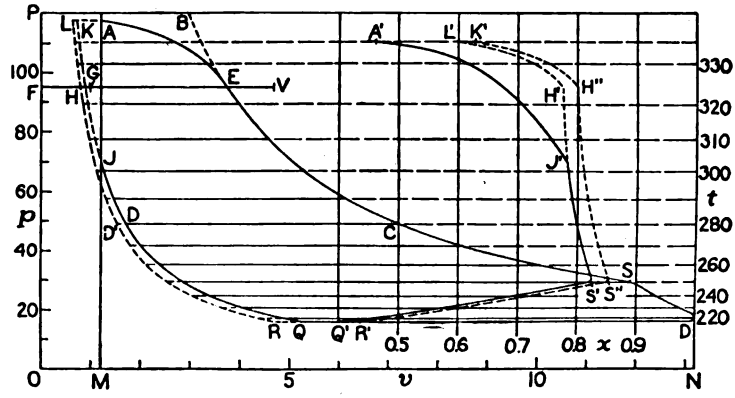

FIG. 117. — Ratio Curves for the Working Steam.

(*i*) RATIO CURVES FOR THE WORKING STEAM. — In Fig. 117 the emphasized quantity is the volume v at any pressure p, as it would enter into the work element $v\,dp$ — see Fig. 67. This v is taken to be simply the width across the diagram, or the intercept between curves AJQ and AESD, and is therefore what we may call the realized or effective volume of the working steam. Continuing our numerical example, the weight of the latter steam is 0.794 lb.; then to get any point on the ratio curve A'J'S'Q', the specific volume s from Table II is multiplied by this weight, and the result is divided into the measured v. At $t = 280$ deg., for instance, $s = 8.64$ and $0.794s = 6.88$ cu. ft.; the volume v or length CD is 5.49 cu. ft.; then $x = 5.49 \div 6.86 = 0.800$

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is the abscissa of the point on curve J'S'. The name "steam ratio" seems more appropriate than "quality fraction" for this x.

In Fig. 117, the compression curve is produced upward to the line of initial pressure, as in Fig. 115; and the expansion curve for this weight of steam is also drawn, so as to complete the cycle diagram of the clearance steam, after the manner of Fig. 111. In this particular case, G is a point on a constant steam-weight curve from Q, like QRT in Fig. 115, and represents the full, dry volume of the clearance steam; similarly, V is a point on the one-pound curve for the total steam, there CC. The point H is located so as to satisfy the proportion

$$FH : FG : : FE : FV.$$

The curve LHR is passed through H and made similar to BS.

The dotted ratio curves are differentiated by the origin of v, at its

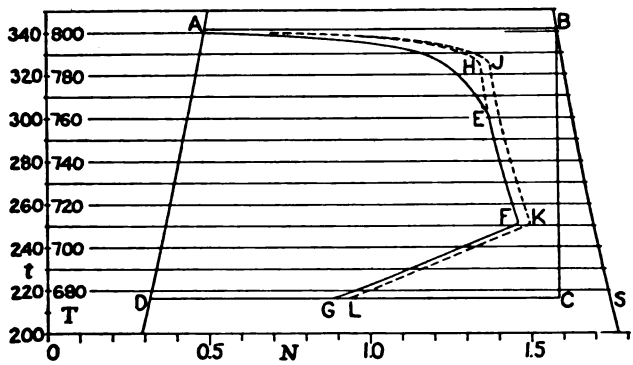

FIG. 118. — Temperature-Entropy Diagram for the Working Steam.

left-hand end on the diagram, the right-hand end being always on the curve AESD. Curve A'J'S'Q' corresponds to AJQ, L'H'S'Q' to KJQ, and K'H''S''R' to LHR. With ratios read from these curves and applied to the vaporization entropy b, the several curves in Fig. 118 are easily plotted. Note that the x curves in Fig. 117 are left incomplete at the top: they would all run over to their zero line at MA.

(j) THERMAL DIAGRAM FOR THE WORKING STEAM. — The full outline AEFGD, Fig. 118, represents the indicator diagram AESDQJ, Fig. 117, transferred on a strictly volumetric basis to the temperature-entropy plane, and there backed up against the water line AD. The losses of effect, from all causes, are shown in their combined value by the departure from the Rankine cycle ABCD; but there is no exact delineation of their manner of occurrence, nor can a detailed analysis be made from this diagram.

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The dotted curves show an approximate analysis of the losses, separating them as regards the agency of their production. Thus if the clearance steam was simply an elastic cushion, its volume always extending to the curve LHR, Fig. 117, and if there were no kinetic waste in filling the clearance space, the working steam would do the work represented by the *pv* area LESDRL or the *TN* area AJKLDA. To the negative clearance-steam cycle directly, but to cylinder-wall action ultimately, is due the lost area between AJKL and AHFG, Fig. 118, while the kinetic waste in filling the clearance cuts off the area AEHA.

In favor of this simple but approximate diagram AEFGD, as against schemes typified by the outline AESDQJ on Fig. 116, the following points may be made:

The curve QJA, Fig. 116, — although as referred to QT or LK it represents fairly well the behavior and effect of the clearance steam, — has no direct thermal meaning; and in being located near the left end of the diagram, in the region belonging to the early stages of vaporization, it entirely contravenes the physical meaning of the entropy abscissas.

The clearance steam has absolutely no relation to the early part of vaporization, but is one of the unfavorable surroundings which, after full vaporization in the boiler, the steam encounters within the engine; it may be considered as merely an agency through which the cylinder walls exert a part of their harmful thermal effect. So far as these influences have weight, the proper place to show their effect is at the right-hand end of the diagram.

In a multiple-expansion engine, with different weights of clearance steam in the successive cylinders, any method like that of Fig. 116 is very troublesome. But diagrams for the unit of working steam alone can be run right down the series of cylinders, with a common reference line in the curve AD.

For closer analysis, whether of a simple or compound engine, curves like ES and Q′J′ of Fig. 116 are to be laid out in true form for each cylinder, along with the diagram like Fig. 118.

With either form of diagram, heat received from condensation of steam in a jacket can very simply be represented by an area to the right of a line like BC in Fig. 118.

§ 25. Thermal Action of the Cylinder Walls

(*a*) INFORMATION AVAILABLE. — The effect of the cylinder walls, as expressed and measured by the size of the missing steam quantity, has been quite fully considered in § 22 — this resultant generally including also a leakage component, which may vary widely in relative magni-

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tude. In the last section, methods of calculating and of graphically representing the wall effect in thermal measure have been set forth, but without the presentation of definite numerical results. Now the action of the metal surfaces will be studied more in detail: the controlling principles can be deduced or inferred, at least in a qualitative way, from the small amount of direct experimental data and from the relations developed in the last three sections.

The essential feature of the experimental investigation is the measurement of the rapidly varying temperature of the metal, and incidentally of the steam also. In its requirement of special skill in the use of delicate apparatus, this line of investigation far transcends the ordinary engineering experiment, lying in the domain of the trained physicist. The instrument available for strictly localized determination is the thermocouple, while the platinum resistance thermometer may equally well be used in the steam. The general scheme is to close the electric circuit for a short and very definitely located portion of the revolution of the crank shaft, say for as little as one-thirtieth, and get the mean temperature during this interval. Since the intervals must be measured serially, quite a time is required for covering the cycle, during which the running conditions must be kept very steady.

Two prominent sets of experiments fully represent the data now available, and comprise the greater part of this body of information. They are those of Callendar and Nicholson, made at McGill University in 1895, Proc. Inst. C. E., 1898, vol. 131, pages 147 to 268; and of Duchesne, at the University of Liège from 1904 on, best reported in *Revue de Mécanique*, 1906, vol. 19, pages 1 to 40.

(*b*) THE STEAM-TEMPERATURE CYCLE. — If the steam is and remains saturated throughout the cycle, its temperature curve can readily

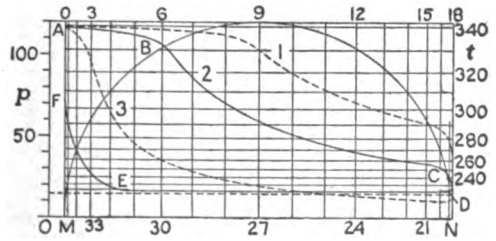

FIG. 119. — Typical Indicator Diagrams, prepared for plot of temperature curves.

be derived from the indicator diagram. In Fig. 119 the three diagrams given cover a wide range of power development, showing a steam distribution of the constant-compression type, as by the Corliss or the double-valve gear. The semicircle on the stroke line MN (which

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merely happens to touch the top line) is divided at every ten degrees of crank angle; then ordinates through the division points locate the corresponding piston positions, as for an engine with "infinite" connecting rod — see § 31 (*b*) and (*d*) for explanation of the relation involved, and for the modification required when the actual motion of the piston is to be more closely followed. As indicated by the scale at the right, horizontal lines are drawn at the pressure heights corresponding to each ten degrees of steam temperature. It is now a very simple matter to plot the saturation-temperature curves in Fig. 120, where the full-line curve No. 2 has the same lettering as its steam diagram.

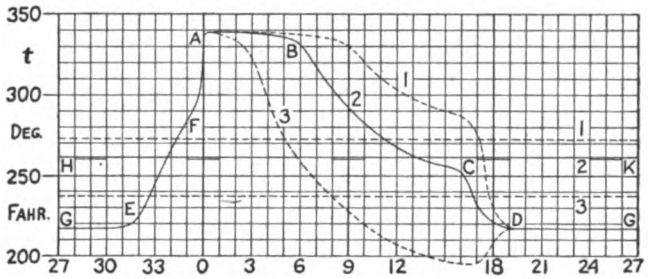

FIG. 120. — Curves of Steam Temperature, from Fig. 119, on the developed crank circle as base.

With these curves are drawn the three lines of mean temperature, type HK, as averaged on the time base of the diagram. An important fact brought out by the experiments is that the mean wall temperature is usually higher than this mean steam temperature.

(*c*) THE CALLENDAR AND NICHOLSON EXPERIMENTS. — For particulars of the engine tested, see beneath Fig. 106. In the head and along the side of the cylinder, flat-bottomed holes were drilled to within a short distance of the inner surface, and the temperature of the surface at the bottom of the hole was measured. The thickness of metal left, or the depth from within outward, was made as little as 0.01 in., but most of the measurements were at 0.04 in. depth. The range of the metal cycle was found to be surprisingly low: in the cylinder head it was, as an average of seven cycles, 4.3 deg. fahr. at 0.04 in., from which was computed a range of 7.1 deg. at the surface exposed to steam. In the same seven runs, the average maximum steam temperature (during admission) was 327 deg., the minimum about 212 deg., and the mean wall temperature 303 deg. On the side of the cylinder the wall temperature was lower, the mean varying from about 292 deg. near the head to 240 at the limit of piston travel (the engine was run single-acting), and the range of fluctuation was greater. As noted on Fig.

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106, the speed of the engine was from 44 to 97 r.p.m., and the temperature range varied somewhat with the speed, as is shown in Fig. 124.

Sample curves are given in Fig. 121, the one marked SS being derived from the indicator diagram as in Fig. 120. The metal curves have the respective ranges 4.9 and 13.5 deg. at the depths named, from which are deduced surface ranges of 7.5 and 20 deg. The latter was the greatest observed, and its size may have been due to the proximity of the narrow, annular space between the inward-projecting head and the counterbore, which had a length of 3 in. parallel to the axis of the

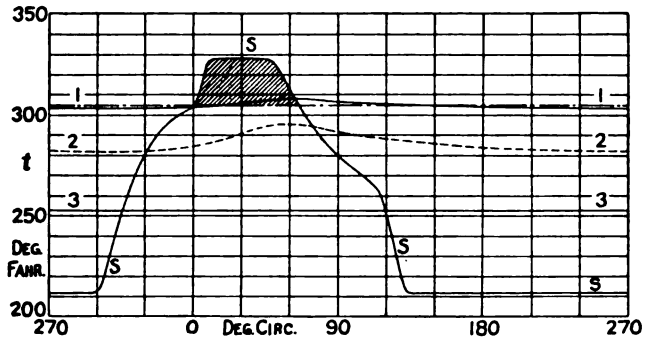

FIG. 121. — Callendar and Nicholson Curves: SS, steam temperature; 1—1, cylinder head at 0.039 in. depth; 2—2, side wall just inside of face of head, at 0.037 in depth; 3—3 mean steam temperature. Speed 44 r.p.m.; test No. 1 in Fig. 106.

cylinder. Perhaps the cleaner condition of the rubbing surfaces will account for the fact that greater ranges were generally observed there than in the cylinder head.

(*d*) CALCULATION OF HEAT ABSORBED. — By methods of mathematical physics, of which only the barest outline is appropriate here, the amount of heat alternately absorbed and given off by the metal wall may be approximately computed when the range of temperature at a known depth has been measured. If the temperature in the indefinitely thin surface layer varies periodically in a definite manner, there will be a similar cycle in the deeper layers: the amplitude of the variation decreases with the depth, and the period lags more and more behind that at the surface. The only form of cycle calculable with any degree of ease is the simple harmonic, in which the temperature t follows a sine curve in terms of time, the complete period 2π being equal to the time of one revolution of the engine. Compare the inertia-force curves of this form in Fig. 192, also "harmonic motion" in § 31 (*b*). With this cycle the temperature range at increasing depths lies between limit curves such as AC and BC in Fig. 122. These are of

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the form $t = e^{-mx}$, where x is the depth from surface AB, according to the scale at the bottom, and m is a function of the specific heat and conductivity of the metal and of the revolution period $1/n$ seconds. Since depth measures volume, hence heat capacity, the area between the curves, as ACB, is proportional to heat absorbed. Since, however, the temperature along the limit curves like AC and BC is not reached simultaneously, but in sequence by the successive layers, the whole heat measured by the area ACB does not go in and out, a portion of it simply flowing back and forth within the metal. The factor for the heat which really passes the surface AB is 0.707 or $\sqrt{\frac{1}{2}}$.

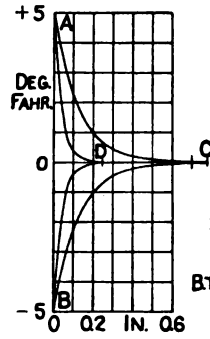

FIG. 122. — Limit Curves of temperature range: ACB for 25 r.p.m., ADB for 250 r.p.m. Simple harmonic cycle.

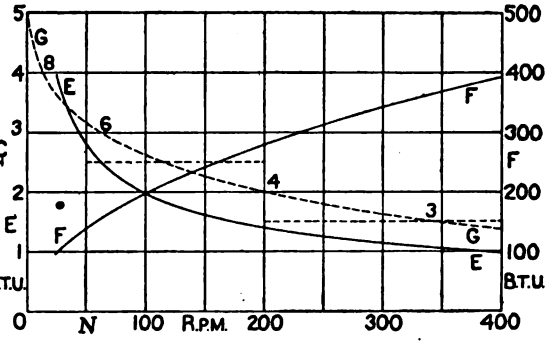

FIG. 123. — Curves of Heat Absorbed, per square foot of surface and with a surface range of 10 deg. fahr.; EE, heat per cycle, FF heat per minute. Curve GG, temperature range at 0.04 in. depth.

Directly applicable results of the calculation are given in Fig. 123: curve EE shows an absorption per cycle per square foot which varies from 4 down to 1 B.t.u. with a change from 25 to 400 r.p.m., while FF shows a total transfer, in each of the two directions, amounting to from 100 to 400 B.t.u. per square foot per minute. These are for a surface range of 10 deg., and will vary in direct ratio to any other range. The ordinate of EE is inversely proportional to $\sqrt{N}$; and when it is multiplied by N to give the heat per minute, this makes the latter quantity vary directly as $\sqrt{N}$. Curve GG shows the calculated range at 0.04 in. depth, with a surface range of 10 deg. as related to speed N.

(*e*) INFLUENCE OF SPEED. — It is of interest to see, in a general way, how the showing of Fig. 123 (for a constant surface range) will be modified by the variation of actual range with speed. For the experiments under consideration, the latter relation is set forth in Fig. 124. Approximately, the range is inversely proportional to some low power of N, or to N^a when a is a small fraction. Then the actual heat ab-

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sorbed per square foot per cycle, which corresponds directly with the steam condensed per cycle, will vary inversely as $N^{0.5+a}$, the heat per minute directly as $N^{0.5-a}$. The missing steam on account of leakage is

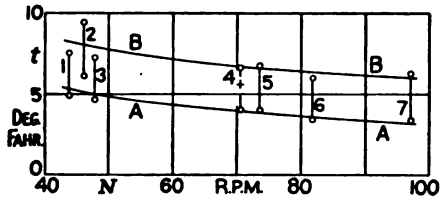

FIG. 124. — Variation of Temperature Range with Speed, in the metal of the cylinder head. Curve AA, observed range t at 0.04 in. depth; BB, calculated range at the surface; odd point marked on ordinate 4 shows one determination at 0.013 in. depth.

also some inverse function of N, so that the general tendency is to make the total missing quantity vary as $\frac{1}{N^{0.5+a+b}}$, while the amount per minute will be proportional to $N^{0.5-a-b}$. In other words, the actual quantities vary more rapidly than the ordinate of curve EE, less rapidly than that of FF.

At first sight there seems to be a serious disagreement between this conclusion and the divisor $\sqrt[3]{N}$, as found empirically for use in Eq. (126); but the less rapid variation of the missing fraction m with speed — because N has a smaller exponent in the formula — is compensated by a change in the total steam received per revolution; which increases as the speed falls off, other conditions remaining the same, because of greater initial condensation. An attempt to bring the matter of the last paragraph to definite, quantitative expression, with the data before us, would be futile and fruitless; but a comparison of Fig. 100 with Fig. 123 shows a close similarity in the manner of variation of the steam condensed and of the heat absorbed per cycle.

(*f*) CALCULATION FROM THE STEAM-TEMPERATURE CURVE. — As appears from Fig. 121, the actual temperature cycle in the metal is not of the simple harmonic form. For an extreme case of concentration of the rapid fluctuation into a short portion of the total period, the authors of the paper give the result of calculations upon an assumed cycle; they state that while the surface range, as deduced from measurements at say 0.04 in. will not agree with the relation represented in Fig. 123, the heat absorbed will be nearly the same. Using then the quantities computed for the harmonic cycle, they establish the following simple method:

In Fig. 121, the shaded area at the top of the diagram SS shows by how much and for how long the temperature of the steam is above that

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of the metal. Only during this period will heat pass from the steam into the metal; and the rate of transfer is taken to be proportional to the difference of temperature — which is only an approximate assumption. Then the product of temperature difference in degrees by time in seconds, or the shaded "condensation area" expressed in degree-seconds, will be a measure of the amount of heat given up by the steam to the metal. With the small ranges observed in these experiments, the line of mean wall temperature may very effectively be used as the lower boundary of the condensation area.

By comparing values of this area with the amount of heat absorbed by the metal according to curve EE of Fig. 123, a fairly constant ratio was found to exist; the mean value obtained for the factor by which to multiply area in order to get B.t.u. was 0.60 for the more regular cycles in the cylinder head, with a range from 0.53 to 0.67. An example will best show the method of calculation.

As originally drawn, Fig. 121 had the scales 1 in. of base = 40 deg. of angle, 1 in. of ordinate = 32 deg. of temperature. The speed being 43.8 r.p.m., the time of one revolution is $60 \div 43.8 = 1.37$ seconds, of which one-ninth (or 40°) is 0.152 sec. Then one square inch of diagram area equals $32 \times 0.152 = 4.86$ degree-seconds. The measured area above curve 1-1 in Fig. 121 was 0.88 sq. in., equivalent to 4.28 deg. sec.; and multiplying this by 0.6 we get the heat absorption 2.57 B.t.u Taking from curve EE, Fig. 123, the rate 3.00 at 43.8 r.p.m., and using the factor 0.75 because the range at the surface is 7.5 instead of 10 deg., we have 2.25 B.t.u. as the heat absorbed by the metal per square foot per cycle. The agreement between 2.57 and 2.25 is not very good, but this happens to be one of the extreme cases, that in which the factor for condensation area worked out 0.53 instead of 0.60.

This scheme is attractively simple, especially if the easily-measured mean wall temperature may be used as the lower limit of the condensation area. It fails, however, to check up well, even over the comparatively small range of these experiments. Thus, if the growth of heat absorption with decrease of speed is to be accounted for, the difference between maximum steam temperature and mean wall temperature ought to become greater with lower speed; actually, for the seven tests plotted in Fig. 124, the difference is practically constant at 24.5 deg., with an irregular variation from 23 to 26 deg.

(*g*) RATES OF HEAT ABSORPTION. — In these experiments, the rate of heat absorption per square foot per minute ranged from 100 to 130 B.t.u. for the surfaces which were, on the average, about 25 deg. below the steam temperature during admission (this 25 deg. being the height of the shaded "condensation area" in Fig. 121); other parts of the

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interior surfaces had higher rates. This is equivalent to a transfer of 4 to 5 B.t.u. per minute per degree of difference between steam and metal; but if we allow for the fact that the absorption took place during about one-fifth of the revolution, the corresponding steady rate would be from 20 to 25 B.t.u. per square foot per minute per degree of difference.

The latter measure of the rate is deduced because of a desire to compare it with the performance of the tubes in a surface condenser. A very complete collection of the information existing on the subject of heat transfer in the condenser will be found in a paper by Mr. G. A. Orrok in Journal A. S. M. E. for Nov., 1910, vol. 32 of the Transactions. In his Fig. 9 are shown rates ranging from 400 to 1000 B.t.u. per hour per square foot per degree of difference between the steam on one side and the water on the other side of a thin wall of brass, the rapidity of the water circulation being the controlling variable: the corresponding rate per minute is 7 to 16 B.t.u. To attempt a close comparison of the physical conditions in the two cases is not worth while, but it is of interest to note that the rather low rates of cylinder-wall absorption deduced above are considerably greater than those maintainable in the condenser.

(*h*) Rates of Condensation. — The Callendar and Nicholson experiments and methods of calculation have been quite fully described, because they represent an almost unique attempt to measure the heat absorption on the side of the metal, instead of deducing it from the missing steam quantity. That the actual quantitative results are typical seems rather unlikely, because the heat rates which they show would generally come far short of accounting for the apparent condensation, and it is difficult to accept the hypothesis of always prevalent leakage sufficient in amount and regular enough in action to bridge the gap.

In that engine, the amount of heat alternately absorbed and given off by the cylinder walls varied from 6000 to 12,000 B.t.u. per square foot per hour (for the different surfaces), equivalent to a condensation of 7 to 14 lb. of steam at about 90 lb. gage pressure: and the clearance surface was 3.74 sq. ft. Apparently calculated by the condensation-area method, the condensation per hour is given as 39 lb., which is not made to vary with the speed — to be explicit, this accounts for line 17 in Table VI of the paper. The missing steam, represented by the length BD on Fig. 106 and got from MB as determined by taking the difference between the very large leakage AB and the total steam used, ranged from 31 to 57 lb. per hour (with no clear relativity to speed), and gave an average of 43 lb. Referred to the 3.74 sq. ft. of

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clearance surface, the measured missing quantity was from 0.0020 to 0.0042 lb. per square foot per cycle.

A few data from the tests diagrammed in § 22 will give an idea of the comparative magnitude of these rates. The 17 by 30 in. noncondensing engine of Figs. 86 and 87 had a clearance surface of 5.81 sq. ft., and the missing steam per square foot of this surface per cycle was from 0.013 to 0.046 lb., the high value belonging to very low speed: at similar speeds this is equivalent to about four times the Callendar and Nicholson result: In the Willans engine, Fig. 96, the missing quantity on the same basis was from 0.0004 to 0.0038 lb. (this range covering the three different sizes of cylinder), but the speed of 400 r.p.m. must be taken into account when making comparison.

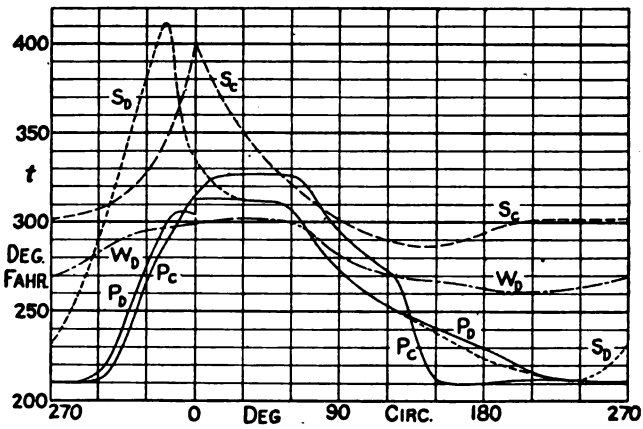

FIG. 125. — Temperature Curves, from observations by Duchesne (subscript D) and by Callendar and Nicholson (subscript C). Curves P, steam temperature from pressure, as in Fig. 120; curves S, steam temperature by thermometer; curve W, temperature of wall surface.

(*i*) THE DUCHESNE EXPERIMENTS, already referred to in Art. (*a*), have been made upon the 300 by 600 mm. experimental engine in the laboratory of the University of Liège, of which the performance under certain conditions has been diagrammed in Figs. 109 to 111. The temperature of the steam was measured by means of a multiple-junction thermocouple, in most of the tests at a distance of 15 mm. (0.6 in.) from the cylinder wall, in some at 1 mm. distance. For the wall, the couple was in the form of a leaf or thin strip of sheet metal, the soldered junction of the silver and platinum halves touching the cast-iron surface, but not adhering in any way. Results from a noncondensing run are given in Fig. 125, replotted from the original diagram, and

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with them a couple of analogous curves obtained by Callendar and Nicholson.

To describe the latter first, the curve S_C shows results got by measuring the steam temperature in a little pocket formed by drilling a $\frac{3}{8}$-in. hole into the thickness of the cylinder head. Strongly affected by the metal in such close proximity, the steam is superheated almost throughout the cycle, and shows a considerable degree of adiabatic rise during compression. But when these experimenters mounted their platinum thermometer upon the piston, in such a fashion as to bring it out into the body of the steam, there was continual practical agreement between pressure temperature and measured temperature. The thermometer was held in the line of the piston axis, with the exposed wire

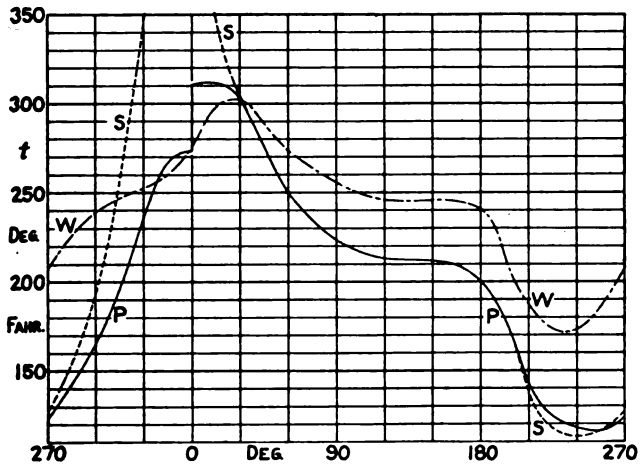

FIG. 126. — Temperature Curves by Duchesne, engine condensing, cut-off at 0.1 of stroke, compression through 0.6 of stroke. For scheme of diagram, see Fig. 125.

at 3 in. distance from the face of the piston. To give room for the projecting thermometer, a capped nipple of 1 in. pipe was screwed into a hole drilled at the middle of the cylinder head. As a consequence of this arrangement, the active part of the thermometer was, during nearly the whole of the compression period, within a narrow space and apparently subjected to a strong wall influence. Unless accounted for, possibly, by the presence of water, the fact that the steam remained saturated is a distinct failure to confirm the Duchesne results.

Turning now to the Liège experiments, the most striking thing seen in Fig. 125 is the superheating of the clearance steam, which begins to appear before the end of exhaust and is very marked during compression. This shows that the latter operation is in large degree adia-

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batic, the walls having become dry before the pressure rise begins, and the rate of heat transfer therefore being low. When condensation is once started, it proceeds very rapidly, and the steam temperature falls to that of saturation. The same kind of action, even more pronounced in magnitude, is shown in Fig. 126, where the S curve which runs off the plane of the drawing has an apex at 525 deg. fahr.

The apparent range of temperature of the wall surface is about 40 deg. in Fig. 125, about 130 deg. in Fig. 126. Whether the thermocouple has always the same temperature as the skin layer of the cast-iron wall which it touches, is certainly a debatable question. There is good reason to believe that the steam will have some influence, the thin leaf of metal taking a temperature somewhere between those of cast

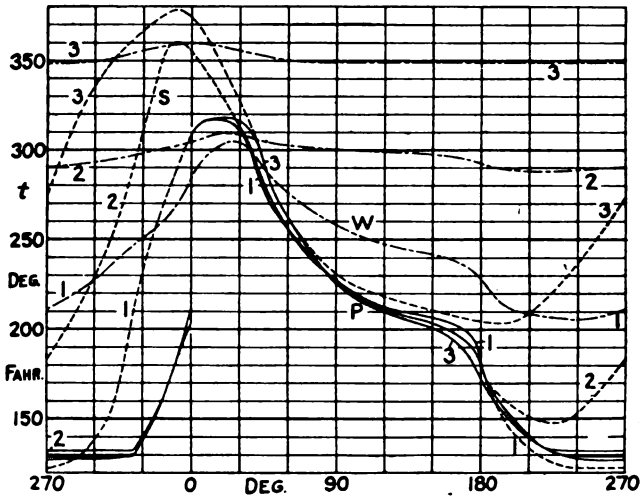

FIG. 127. — Temperature Curves under Various Conditions as to Steam Jacketing.

iron and of steam, although probably much nearer the former. That there may be a considerable variation in the effectiveness of contact is suggested by the fact that the two ranges quoted above have between them a difference much greater than any possible difference between the amounts of heat transferred, or of steam condensed and reëvaporated, in the respective cases. Making every allowance for these uncertainties, there is indubitable indication of a real surface range decidedly greater than was found in the Callendar and Nicholson tests.

Yet another set of these results is given in Fig. 127, of interest here and also in connection with the subject of steam jacketing. The three types of curves are distinguished by the kind of line, as in the preced-

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ing figures. The respective conditions are: case 1, no steam in jackets; case 2, jackets filled with steam of boiler pressure; case 3, jacket steam of higher pressure and temperature (amount not stated). Note the successive elevation of the mean wall temperature, and the accompanying diminution in its range of fluctuation; also how the superheating of the steam during exhaust and compression increases as the wall is hotter.

(*j*) CONCLUSIONS. — It is very evident that the data presented are neither sufficient in amount nor in good enough agreement to be formulable into even an empirical theory of wall action; they do serve, however, to elucidate some important principles.

That the mean temperature of the cylinder walls is generally a good way above the mean of the steam temperature indicates that the freedom of heat transfer from steam to metal is greater than from metal to steam. This is highly reasonable under the conditions of the steam-engine cycle, for when giving up heat the steam is at high pressure, when it is receiving heat the pressure is low and falling. Probably, conductivity increases slowly with density; but the stronger influence is the rising or steady high pressure, which holds the moisture of condensation against the metal; and while water is a poorer conductor than iron, the wetness of the surface promotes the yielding of heat by the vapor. As the pressure falls, the film of water will tend to be thrown off by the formation of steam beneath it; and with a sudden drop of pressure at release, a good part of the water mixed with the steam is likely to be swept out unevaporated. By these mechanical actions, as well as the simple outflow of heat, the layer of steam close to the metal is made dry, and thus the freedom of heat transfer is yet further diminished.

Callendar and Nicholson advance the proposition that as the influences favoring cylinder condensation grow stronger, the mean wall temperature will fall; but that the lower limit is the mean temperature of the steam, as fixed by its pressure. When this limit is reached, water may accumulate in the cylinder; or, at least, moisture will be present throughout the cycle, even though any great accumulation is prevented by the mechanical action of the steam currents. These hypotheses suggest a rational idea of the influence of moisture in the entering steam: if the condensation is less than the limit just described, so that the cylinder walls are dry at the beginning of admission, the degree of wetness of the new steam may exert a considerable influence; but if the cylinder action is strong, and the walls already moist when admission begins, a little more water in the entering steam makes very little difference. This is in accord with experience.

The "condensation-area" method of Callendar and Nicholson can-

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not be considered as satisfactorily established. The general idea may be correct, although subject to secondary influences not yet clearly seen; but the reduction factor deduced from their tests is not to be accepted without confirmation by other and far more extensive measurements of temperature range in the metal. There is certainly no sufficient reason for substituting a scheme of calculation or estimate based upon temperature observations for the more direct determination of (probable) missing quantity by the method of Eq. (126).

A very great practical disadvantage of the whole plan of direct temperature measurement, as a means of investigation, is found in the non-uniformity of condition within the cylinder; whence observations so strictly localized must be made at a number of different points, in order that the prevailing mean may be ascertained. Against this, the indicator diagram gives a net result, perhaps of very heterogeneous conditions, but all the more useful because it is their net effect.

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CHAPTER VI

PERFORMANCE AND EFFICIENCY OF THE ENGINE

§ 26. Measures of Performance

(*a*) STEAM CONSUMPTION. — The most obvious and the most generally used measure of the thermodynamic performance of an engine (or of a turbine) is the steam consumption per unit of work output. For the piston engine, the latter unit is commonly the horse-power-hour; and because the indicator offers the easiest means of measuring engine output, the indicated power, at the piston, is much oftener used than the effective or "brake" power. In the service of driving electric generators, the electrical output in kilowatts is both more important and more readily known than the steam horse-power; with the turbine, in fact, the latter can be found only by inference, through an estimate of machine losses. For these reasons, steam per kilowatt-hour is a much used measure of performance. The relative values of the several output units are as follows:

1 horse-power-hour = 1,980,000 ft. lb. = 2545 B.t.u.
= 0.746 kilowatt-hour.

1 metric horse-power-hour = 270,000 mkg. = 0.9863 English h.p.h.
= 0.736 kw.h.

1 kilowatt-hour = 1.3405 Eng. h.p.h. = 3412 B.t.u.

For the steam rate, the symbol S will be used. When distinction is necessary, S_h will be pounds per horse-power-hour, S_k pounds per kilowatt-hour. The relations are,

1 lb. per h.p.h. = 1.340 lb. per kw.h.
= 0.4474 kg. per met. h.p.h.

1 lb. per kw.h. = 0.746 lb. per Eng. h.p.h.
= 0.4536 kg. per kw.h.

1 kg. per met. h.p.h. = 2.235 per Eng. h.p.h.
= 1.359 kg. per kw.h.

(*b*) ACTUAL STEAM RATES. — The consumption per indicated horse-power-hour, or S_h, varies from more than 100 lb. in small steam pumps down to just about 9 lb. in the best engines and turbines with highly

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superheated steam. For several important classes of engines, good average performance, with saturated steam, is about as follows:

Small noncondensing engines	30 lb.
Large noncondensing engines	25 "
Locomotives	24 "
The medium range of condensing engines	17 "
Big and well kept power engines	13 "
The best pumping engines	11 "

Only among engines of the same general class, especially as to pressure limits and as to kind of steam used, is the consumption S a good standard of comparison. For the plant as a whole, and with emphasis upon the idea of efficiency, this measure may be quite misleading, because of wide variation in the amount of heat required, from the fire, to make one pound of steam. Thus the locomotive, almost never supplied with a feed-water heater, must use cold feed water; and in winter its pound of high-pressure steam may require 180 B.t.u. more than the same weight in a stationary noncondensing plant with a good exhaust-steam heater. Again, $S = 9$ in a high-superheat engine is little, if any, better than $S = 11$ with saturated steam. And as an extreme case, engines with the regenerative cycle of Fig. 59 use more pounds of the same kind of steam than do those with the common cycle, for the same or better efficiency. Refer to Table 13, page 268, and compare tests Nos. 39 and 40 with the just preceding results from ordinary pumping engines.

(*c*) THERMODYNAMIC EFFICIENCY. — The work done per pound of steam, preferably expressed in heat units, so that the symbol would be AU in the terms of Chapter IV, and now represented by W, is

$$W = \frac{2545}{S_h} = \frac{3412}{S_k} \text{ B.t.u.} \quad . \quad . \quad . \quad . \quad . \quad (136)$$

as already used in § 22 (*e*). With S_h ranging from 30 lb. down to 10 lb., W will lie between 80 and 250 B.t.u. Preparing now for an extensive application of the ideas set forth in § 8 (*g*), we see that W is useful for comparison with the heat Q required from the fire to make one pound of steam, in order to get the absolute thermal efficiency

$$E_A = \frac{W}{Q}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (137)$$

also, that by comparing it with the output W_R of the unit of steam in the ideal (Rankine) cycle, computed as in § 15 (*d*), is obtained the relative efficiency

$$E_R = \frac{W}{W_R}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (138)$$

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In THE STEAM ENGINE, Vol. II, Chapter XIII, are given tables showing results from tests of more than 110 engines, in all of which the efficiencies just defined are worked out. Some of the more prominent of these tests are discussed in the next section, and set forth in Table 13. A general idea of realized results, as to absolute efficiency, can be got from the following summary:

Type of Engine.	Efficiency.
Simple engines	0.07 to 0.13
Locomotives	0.09 to 0.13
Medium and large power-service engines, including cases with moderate superheat	0.12 to 0.18
The best power engines	0.18 to 0.20
Modern high-superheat compounds	0.19 to 0.21
High-grade pumping engines	0.18 to 0.22
Good average marine engines	0.15 to 0.17

The relative efficiency ranges mostly from 0.60 to 0.67, sometimes rising above 0.70 and seldom falling below 0.50 except when the conditions of working are very bad, or the engine is much underloaded or overloaded. The upper limit is about 0.75 in condensing engines and 0.80 in noncondensing — see the column for E_R in Table 13. These limits are approached by only the best engines, large and in good condition. The application of relative efficiency as a criterion of performance is more fully discussed in Art. (*k*).

(*d*) HEAT IN THE FEED WATER. — In determining the amount of heat to be charged as input Q, several questions arise. If the purpose is to find the efficiency of the whole plant, the obvious and proper thing is to use both feed temperature and steam quality as measured near the boiler: applying these as in § 13 (*d*), we get the heat actually received from the fire. But if the engine is to be judged by itself, and if comparisons are to be made among engines, losses due to the other elements of the plant must be eliminated. In the ideal cycle, § 15 (*d*), the temperature of the feed water is that of the steam exhausting from the engine. The open feed-water heater, in which the water falls in spray through an atmosphere of exhaust steam, will deliver water within five degrees of the exhaust temperature, if not worked too rapidly: but so far as the exhaust from the main engine is concerned, this device is available only in a noncondensing plant. The surface heater will generally show a larger gap, say from ten degrees upward, according to the rate of water flow relative to the amount of heating surface. Allowing for some loss from the feed pipe, often not well covered, the heat q_0 in the feed water is likely to be from 5 to 15 B.t.u.

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less than the value corresponding to the exhaust temperature. From this best operation of the heating system, conditions range downward, through all degrees of low effectiveness, to the feeding of cold water.

It would be most logical to establish or assume an average deficiency in the heat q_0, saying that with a good feed-water system this would be so many heat units less than at exhaust temperature, and thus including the unavoidable loss in the working of the engine. Uncertainty as to the proper allowance makes the following scheme more desirable:

To get the engine efficiency alone, use an ideal temperature t_0 for the feed water; this will be either the actual temperature of the steam in the exhaust pipe, or with jackets, as described below, some higher temperature.

For boiler efficiency, use the temperature in the feed pipe taken near the boiler, beyond the feed heater and pump. When the boiler is fed by an injector, the low temperature in the suction pipe of the latter will generally be taken as the feed temperature — see the elaboration of this matter in Art. (*h*). An economizer is to be considered as a part of the boiler, since the heat which it saves is a part of the heat of combustion.

(*e*) IDEAL FEED TEMPERATURE. — When the engine is provided with steam jackets or reheater coils, in which water collects by condensation but need not fall below the temperature of the steam from which it is condensed, the ideal action would be the return of this water to the boiler without the loss of any of its heat. If it can be mixed with the main current of feed water, coming from the perfect heater at exhaust temperature, the result will be the limiting feed temperature for the particular engine. Suppose, for instance, that in an engine which receives steam at 120 lb. abs. the amount condensed in the jackets is 10 per cent of the total, all the jackets carrying full pressure, and that the exhaust temperature is 110 deg. At $p = 120$, $t = 341$ and $q = 312$; at 110 deg. q is 78. Then 0.1 lb. of hot water from the jackets will contain $0.1 \times (312 - 78) = 23.4$ B.t.u. above 110 deg., and will raise the temperature of the whole pound (including itself) to 133.4 deg., which is therefore the ideal temperature in this case.

(*f*) VARIOUS EFFICIENCIES. — When Eq. (137) is to be applied to the plant as a whole, all the heat Q_b absorbed from the fire by the boiler, per unit of engine output, must be charged against that output. For the engine alone, only the heat Q_e carried by the steam which goes into the engine should be charged. The difference between Q_b and Q_e represents heat which is used up in driving the auxiliaries, dissipated in steam-pipe losses, or required to make up deficiencies in

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the feed-heating system: its apportionment among these objects may be a matter of considerable complexity.

The efficiency of the engine, when calculated with the ideal feed temperature and with steam of the quantity and quality delivered by the separator, is the limit of attainable plant efficiency, more nearly approached as the pipe losses are less, the feed heater more effective, and the auxiliaries more economically operated. Before illustrating the calculation of efficiencies and the distribution of losses, it will be necessary to consider more fully the behavior and the influence of the minor members of the plant.

(*g*) Working of the Auxiliaries. — These consist of the feed pump, the condenser pump or pumps, and sometimes special pumps for returning jacket water, circulating oil, etc. With very uniform loading, as in a pumping-engine plant, they may be coupled to the main engine, and then consume a small portion of its output of power; in some modern steam-electric plants, the auxiliaries are driven by motors, again drawing upon the main output; but by far the commonest scheme is to use separate steam-driven pumps. In developing from 2 to 5 per cent of the power of the engine, in a condensing plant, the pumps are likely to use from 5 to 20 per cent of the total steam consumed — the larger ratio belonging to smaller plants and accompanying poorer conditions in design and maintenance.

A minimum in the thermodynamic cost of operating the auxiliaries is attained when their exhaust, all of it if possible, is used to heat the feed water. Whether by drawing from the hot well or by having a tubular heater in the exhaust line of the engine, a good approach to the temperature of the exhaust steam can be made. Between the temperature thus fixed by the main engine and the maximum of 212 deg. for water under atmospheric pressure there is room for the absorption of a considerable amount of heat, perhaps 100 B.t.u. If the exhaust from the pumps does not carry more than this much available heat per pound of feed water, it can all be condensed by the latter and either taken into the main current (if properly freed from cylinder oil), or at least thrown away as water but little hotter than the final feed. If this return or "regeneration" of the heat supplied to the auxiliaries is complete, the operation of the latter costs almost nothing, or merely the insignificant heat equivalent of their useful work.

(*h*) Feed-water Data. — How to fix upon the proper weight and temperature of the feed water, in the scheme of working just outlined, can be made clearer by considering the closely analogous case of feeding the boiler by an injector — the action of the latter being described in § 52. Per horse-power-hour of the engine, the injector draws w_1 lb. of

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water at t_1 deg.; and by the admixture of live steam increases the weight to w_2 and raises the temperature to t_2. Barring radiation loss and the little bit of work done in pushing the water into the boiler, all the heat of the steam supplied to the injector is returned as heat. The effect is that of a live-steam feed-water heater. So far as the amount of heat delivered to the engine, from the fire, is concerned, it is immaterial whether we take w_1 lb. of steam from water at t_1 or w_2 lb. from water at t_2. The latter represents more truly the manner in which the heat is received from the fire; but the former is decidedly the proper form of quantity to be used with the engine. The net feed (w_1 lb.) enters into the main cycle of the plant; the increase ($w_2 - w_1$) runs parallel with w_1 for a little way, then turns off into the secondary cycle, between boiler and injector, in which it keeps up a continual circulation.

The application of the ideas set forth in this and the preceding articles can best be shown by means of an example. To formulate rules covering all the cases which are likely to be met with in steam plants would be a laborious and space-filling task. Sometimes, conditions are very simple; very often a number of secondary quantities are involved, more or less difficult to measure, some of them perhaps derivable by inference. A clear understanding of principles, with some ingenuity, is the proper equipment for such work.

EXAMPLE 31. — In a pumping-engine plant, cold water at 52 deg. is drawn through a surface heater located between engine and condenser, and raised to 102 deg. by heat from the main exhaust; it then passes to an open heater, where the exhaust from the auxiliaries and the hot water drained from the jackets and separator combine with it to form a mixture of 205 deg. temperature. At the boiler the steam has a pressure of 162 lb. abs., and contains 0.6 per cent of moisture; at the engine, beyond the separator in the steam pipe, the pressure is 158 lb. abs. and the quality 0.995.

In a test of this plant, the jacket and separator drains were cut off from the heater; the several discharges were passed through calibrated measuring vessels, and then allowed to run to waste. The open heater now produced a temperature of 168 deg., and the feed water was weighed between this heater and the pump. The following quantities were determined:

Weighed feed water	5240 lb. per hour.
Drained from the engine separator	66 " " "
Measured jacket water	746 " " "
Drained from oil and water separator of open heater	124 " " "
Indicated power of engine	448 i.h.p.

Analyze this test for steam rates, efficiencies, etc.

A. Determine the steam consumed by the auxiliaries.

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Since the exhaust from the pumps passes through a separator before it enters the open heater, we assume it to be made nearly dry at atmospheric pressure, taking 1140 B.t.u. as its total heat. The hourly weight x lb. of this steam may be thought of as condensed and cooled to 102 deg.; then the $1070x$ heat units which it thus gives off suffice to raise the whole 5240 lb. of feed (including the condensed steam) from 102 to 168 deg. By the equation

$$1070x = 5240 \times 66,$$

we find x to be 323 lb. Adding the oil-separator drainage, the total consumption by the pumps is 447 lb.

B. Check the jacket and steam-separator discharge, of 812 lb. weight.

It is reasonable to assume that the consumption of steam by the engine, in whole and in parts, will not be modified by the change in arrangements found necessary for weighing the feed water. Then only $5240 - 812 = 4428$ would normally come from the supply and the pump exhaust; and the heat given up by the latter, still $1070 \times 323 = 345{,}600$ B.t.u., would raise this weight by

$$345{,}600 \div 4428 = 78.0 \text{ deg.},$$

or from 102 to 180 deg.

The raising of the whole 5240 lb. of water from 180 to 205 deg. requires $5240 \times 25 = 131{,}000$ B.t.u.; so that above 180 deg. each pound of hot water returned must give off, in the ordinary working of the plant,

$$131{,}000 \div 812 = 161 \text{ B.t.u.}$$

At the steam pressure of 158 lb. abs., the temperature is 363 deg. and the heat of the liquid 334.5 B.t.u.: between 334.5 and $(180 - 32)$ there is a difference of 186.5 B.t.u. which might be carried, as available heat, by the pound of water from the high-pressure system. Comparing this with the 161 B.t.u. actually delivered, we see the effect of a reasonable loss by radiation plus that of the use of steam of reduced pressure (and temperature) in some of the jackets.

C. The steam (or water) quantities per hour now work out as follows:

Total water pumped into boiler	5240 lb.
Used by auxiliaries	447 "
Sent to main engine	4793 "
Drained from separator	66 "
Used by main engine	4727 "
Used in jackets	746 "
Used in cylinders	3981 "

D. Efficiency of whole plant.

At 162 lb. abs., from water at 205 deg. and with 0.006 of moisture, the heat of formation of one pound of steam is,

$$Q = 1195.1 - 173.0 - 5.1 = 1017 \text{ B.t.u.}$$

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The total steam rate for the plant is,

$$S = 5240 \div 448 = 11.70 \text{ lb.}$$

Then the heat supplied per horse-power-hour is $1017 \times 11.70 = 11{,}900$ B.t.u., and the efficiency is,

$$E = \frac{2545}{11900} = 0.214.$$

E. Effect of the auxiliaries.

The pumps use 447 lb. of steam, which costs, as above, 1017 B.t.u. per pound, and return 323 lb. carrying $1140 - 173 = 967$ B.t.u. per pound, above the normal feed temperature; then the net cost is

$$454{,}600 - 312{,}400 = 142{,}200 \text{ B.t.u.}$$

Per indicated horse-power of the main engine this is equivalent to $142{,}200 \div 448 = 318$ B.t.u. per hour, so that the heat really supplied for the engine is $11{,}900 - 318 = 11{,}582$ B.t.u., and the efficiency of the latter becomes,

$$E = \frac{2545}{11{,}582} = 0.220.$$

F. Limiting efficiency, of the engine alone, with the ideal feed temperature.

In view of what is brought out in division B above, take the effective heat of the jacket discharge to be about 320 B.t.u. per pound (above 32 deg.), or 250 B.t.u. above 102 deg. Per horse-power-hour, the consumption of steam by the engine alone is

$$S = 4727 \div 448 = 10.55 \text{ lb.},$$

of which the fraction used by the jackets is $746 \div 4727 = 0.1578$. The ideal feed temperature is therefore $102 + (0.158 \times 250) = 141.5$ deg. — see Art. (*e*) — and at 158 lb. abs., with $m = 0.005$, the heat of formation is,

$$Q = 1194.7 - 109.5 - 4.3 = 1082 \text{ B.t.u.}$$

With a heat supply of $10.55 \times 1082 = 11{,}420$ B.t.u., the efficiency becomes,

$$E = \frac{2545}{11420} = 0.223.$$

The only element that has any weight in making this E differ from the value 0.220, found above without the auxiliaries, is the steam-pipe loss.

(*i*) HEAT CONSUMPTION. — The heat supply per unit of output is a better basis of judgment, and more directly related to thermal efficiency, than the steam rate. Especially, to give the amount of heat supplied per horse-power-hour makes the steam engine readily comparable with the gas engine, since that is the most commonly used measure of performance for the latter. The two output units having the values 2545 B.t.u. for the horse-power-hour and 42.42 B.t.u. for the horse-power-minute, the heat rate Q is related to efficiency as shown by Table 11.

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TABLE 11. THERMAL EFFICIENCY AND HEAT SUPPLY.

Efficiency.	Heat per		Efficiency.	Heat per		Efficiency.	Heat per	
	Hour.	Minute.		Hour.	Minute.		Hour.	Minute.
0.05	50,900	848.4	0.12	21,210	353.5	0.19	13,395	223.3
0.06	42,420	707.0	0.13	19,580	326.3	0.20	12,725	212.1
0.07	36,360	606.0	0.14	18,180	303.0	0.21	12,119	202.0
0.08	31,810	530.2	0.15	16,970	282.8	0.22	11,568	192.8
0.09	28,280	471.3	0.16	15,910	265.1	0.23	11,065	184.4
0.10	25,450	424.2	0.17	14,970	249.5	0.24	10,605	176.7
0.11	23,140	385.6	0.18	14,140	235.7	0.25	10,180	169.7

(*j*) DUTY OF PUMPING ENGINES. — For this class of engines a special measure of performance is commonly used, called "duty." In early practice it was defined as the number of foot pounds of work done per 100 lb. of coal burned. Because the heating power of coal is a variable and uncertain quantity, this has been changed to work per 1000 lb. of steam consumed; which is roughly equivalent to the older rate, since one pound of good coal will evaporate about ten pounds of water. For really definite comparison, the better standard is foot pounds of work per million B.t.u. received from the fire — this input quantity being derived from the thousand pounds of steam. These two measures are both in current use, the former often preferred for commercial purposes because it gives a more distinctly separate test of the engine and condenser.

Duty is not determined by indicated steam work, but by the work of the water pistons or plungers, so that it is in terms of net work output. In some cases it has even been based upon actual water delivered, thus including the effect of leakage and "slip" in the pump; but the common practice is to credit the full work of the plungers, making or meeting a separate specification in regard to delivery. The effective pressure pumped against is got by taking the suction lift and the discharge pressure, as determined by gages near the pump, and adding to their sum the pressure equivalent of any difference of level between the gages: this credits the pump with the overcoming of all pipe resistances, in addition to the actual lift from suction level to discharge level, but not with work done against its own internal hydraulic resistances.

With the million B.t.u. base, duty is readily transformed into thermal efficiency. Suppose, for example, that its value is 165 million:

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dividing out the million, we have 165 ft. lb. per one B.t.u. of heat supplied, or 165 of output per 778 of input, so that the efficiency is

$$E = 165 \div 778 = 0.2123. \quad . \quad . \quad . \quad . \quad . \quad (139)$$

For the ordinary type of Rankine-cycle engine, duty per 1000 lb. of steam will range higher than per 1,000,000 B.t.u., because the heat Q per pound of steam is likely to be in the neighborhood of 1080 B.t.u.; with the regenerative cycle, as described in § 27 (*i*), the heat-unit basis gives the higher value, since Q may be made less than 900 B.t.u. The large vertical pumping engine has a high mechanical efficiency, commonly from 0.93 to 0.95, as between indicated steam power and duty power. Good, high values of duty, somewhat better than will be maintained in regular running are

Per million B.t.u., 165,000,000 ft. lb., and

Per 1000 lb. of saturated steam, 180,000,000 ft. lb.

(*k*) RELATIVE EFFICIENCY. — The ratio of actual to ideal output is the true criterion of the performance of an engine, when we are considering how effectively the controllable losses are eliminated or diminished. The logical and proper "ideal" standard is the Rankine cycle; various other "theoretical" diagrams have been proposed, but they can be rated only as makeshifts. There are, however, some drawbacks to the use of the relative efficiency defined by Eq. (138), due chiefly to wide variation in the amount of work lost through incomplete expansion. The example illustrated in Fig. 128 will help to make this matter clear.

In that figure an assumed temperature-entropy diagram for a compound engine is shown in dotted lines. Disregarding the effect of air in the condenser, we take temperature t_0 and pressure p_0 to be in the steam-table relation: then, with release at 8 lb. abs., the exhaust line is varied from 160 deg. to 70 deg., or from 4.7 to 0.4 lb. The derived curves, numbered 1 to 5, are laid out on a vertical base of this exhaust temperature, to the enlarged scale at the right.

Curve 1 (scale at top) shows the actual effective output W, in B.t.u., or the total enclosed area of the two steam diagrams, down to the particular exhaust line: thus for 130 deg., it is the area above DGH. Because the volume of the cylinder is so small relative to the specific volume of the steam at very low pressures, W is scarcely increased at all by lowering t_0 below 100 deg.

Curve 2 (same scale as 1) gives the Rankine-cycle output W_R, plotted from column 2 of Table 6, page 116. This increases with fall of t_0 at a nearly constant rate.

Curve 3 (scale at bottom) is the relative efficiency, $E_R = W \div W_R$.

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Its wide variation with a comparatively small change in the unit output W is what detracts a good deal from its real usefulness as a standard of comparison.

Curve 4 (scale along curve) shows how the heat input $Q = H_1 - q_0$ varies with changing t_0; on the base here used the relation is represented by a straight line.

Curve 5 (scale near top) gives the absolute efficiency $E = W \div Q$. At first this ratio shows a fair rate of increase with vacuum; but it

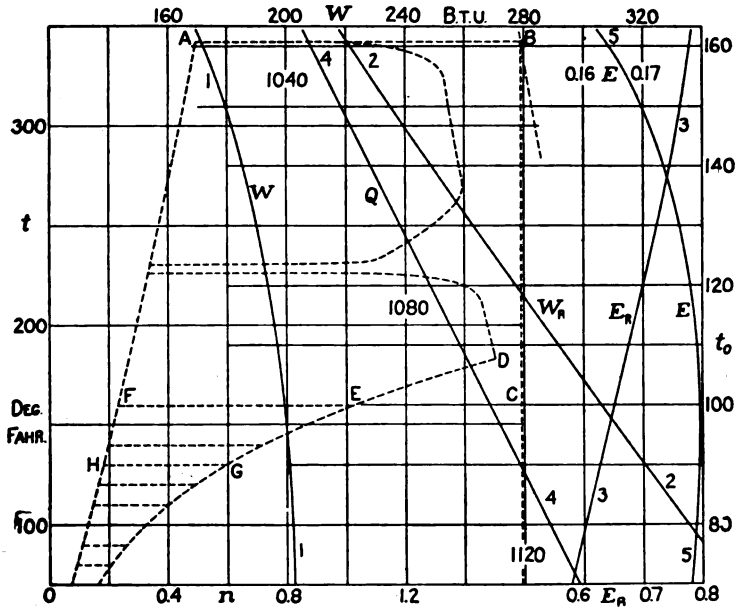

FIG. 128. — Example Showing Effect of Variant Exhaust Pressure, all other conditions remaining unchanged, including the terminal or release pressure. Initial pressure, $p_1 = 120$ lb. abs.; release pressure $p_2 = 8$ lb. abs., equivalent to 183 deg. temperature; quality x_1, with effective cut-off as at p_1, about 0.75; volume per pound at and during release, 38 cu. ft.

reaches a maximum at about $t_0 = 100$ deg., then begins to decrease slowly.

The engine diagram in Fig. 128 is typical in its proportions, especially as to the terminal condition of the steam, at point D — see § 15 (g) for a statement of the considerations which limit the amount of practically advantageous expansion. The unfairness of the relative efficiency, represented in curve 3, as a basis of comparison among engines is apparent from the fact that an engine with a poor vacuum may show a better relative performance than another which is working with a good vacuum, doing more work per pound of steam, and de-

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veloping a higher absolute efficiency. To overcome this difficulty, it has been proposed to stop the adiabatic expansion, in the ideal engine, at a terminal pressure equal to that in the actual engine, then drop to the exhaust pressure by cooling at constant volume — in strict historical accuracy, this modified scheme is the reference cycle favored by Rankine, while Clausius preferred to adhere to the cycle with full expansion. The objection to this plan is that the introduction of another condition makes the ideal operation less definite and determinate, the calculation of its output becoming more troublesome and complicated. Further, for the steam turbine, which is under no such volumetric restriction as the engine, the full-expansion diagram represents decidedly the amount of available work: in the capability of utilizing the triangular area under the curve DEG, Fig. 128, the turbine has its one really strong point of thermodynamic superiority over the engine.

(*l*) A SIMPLE RELATIVE STANDARD. — The trouble with the scheme leading to curve 3, Fig. 128, is that a change in exhaust pressure which has a great effect upon the output of the ideal cycle can produce but a small change in the output of the actual engine. Since practical limitations of cylinder volume thus greatly diminish the

TABLE 12. RANKINE-CYCLE OUTPUT W_R, EXPRESSED IN B.T.U., PER POUND OF STEAM INITIALLY DRY-SATURATED.

Lower Limits,
For noncondensing engines, $p_0 = 15$ lb. abs., $t_0 = 213$ deg.
For condensing engines, $p_0 = 1.94$ lb., $t_0 = 125$ deg.

p_1	Output.		p_1	Output.		p_1	Output.	
	Atm.	Vac.		Atm.	Vac.		Atm.	Vac.
250	203.8	316.8	150	167.8	284.4	75	117.7	239.2
240	201.0	314.3	140	162.8	279.9	70	112.7	234.7
230	198.0	311.6	130	157.5	275.1	65	107.3	229.8
220	194.9	308.8	120	151.8	269.9	60	101.4	224.5
210	191.6	305.9	110	145.5	264.3	55	95.0	218.7
200	188.2	302.8	100	138.5	258.0	50	88.1	212.4
190	184.6	299.5	95	134.8	254.7	45	80.4	205.0
180	180.8	296.0	90	130.9	251.1	40	71.9	197.7
170	176.7	292.4	85	126.8	247.4	35	62.4	198.2
160	172.4	288.6	80	122.4	243.4	30	51.9	179.8

influence of variation in back pressure, the latter almost ceases to be a major determinant of real performance. In view of this fact, a simple and convenient standard of reference may be established by choosing

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mean or typical values of exhaust pressure, for the respective cases of noncondensing and condensing engines, then calculating the Rankine-cycle outputs with these lower limits, as was done in preparing Table 12. The choice must be somewhat arbitrary (in the mathematical sense of the word): the values here used, specified in the heading of the table, are taken as a little better than the average of really good practice, looking rather at the back pressure upon the piston than at the pressure in the condenser. When comparison is made with an ideal performance taken from this table, excessive back pressure will simply be bunched with the other sources of loss. For many purposes, such a comparison, beside being easy to make, will be more satisfactory than one with the result of the relatively exact calculation outlined in Example 15, page 105: the latter "standard" is used, however, in Table 13.

(*m*) Economical Vacuum. — As already noted concerning the typical example in Fig. 128, the absolute efficiency shown by curve 5 has its maximum at about 100 deg., while there is practically no gain below 125 deg. The assumption which underlies this curve — namely, that the back pressure on the piston corresponds (as for pure steam) to the temperature in the condenser, which temperature is retained by the water of condensation — is at once simpler and more favorable to high vacuum than is the actual state of affairs. Because of air in the condenser, always present to some extent, the pressure there is higher than that corresponding to the temperature; and from condenser to engine there is the added back pressure due to pipe and port resistances. In a given engine, with fixed sizes of ports, etc., the difference between the exhaust pressure in the cylinder and the ideal condenser pressure will decrease somewhat as the latter is made lower, but in much less than direct proportion; so that it will have a relatively greater harmful effect as the vacuum is increased. This will swing the lower part of curve 1 toward the left, raising the maximum on curve 5.

The limit of commercial efficiency is farther up the scale of exhaust pressure than that fixed by purely thermodynamic considerations. To get the very high vacua which are found profitable in steam-turbine practice requires a larger and more expensive condenser outfit than can be made profitable with the engine; and if the supply of cooling water is at all limited, it will not pay to use an excessive amount. To maintain an absolute pressure of 0.75 lb. in the condenser will probably cost twice as much, in apparatus and water, as to maintain 1.5 lb.; while 1.5 lb. will cost very little more than 3 lb. The turbine can realize a large proportion of the ideal gain from the higher vacuum, the engine but a small fraction of it. Plant conditions are so varied that no closely quantitative generalization is warranted; but, roughly,

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to make the vacuum greater than 27 inches of mercury, or the absolute condenser pressure less than 1.5 lb., will not produce any net gain in a steam-engine plant. Further description and discussion of condenser action will be found in Chapter XI.

(*n*) EQUIVALENT STEAM RATES. — In expressing performance by the steam rate, it is quite customary to reduce actual consumption to equivalent dry steam. This is done on the basis of equal heat content, above the initial temperature of heat reception (generally the feed-water temperature). Suppose, for instance, that steam at 120 lb. abs., from water at 180 deg., contains 2 per cent of moisture. The heat of formation of dry steam, with this p_1 and t_0, will be 1189.8 − 127.8 = 1062.0 B.t.u., and that of the wet steam, 1062.0 − 17.6 = 1044.4 B.t.u. Then one pound of the latter will be equivalent to 1044.4 ÷ 1062.0 = 0.9843 lb. of the former. If the engine uses 15.6 lb. of steam per horse-power-hour, the equivalent dry steam will be 15.6 × 0.9843 = 15.36 lb. With superheated steam, the rate number will be increased by reduction to the equivalent at dry saturation.

Frequently, instead of getting the heat-bearing equivalent, testing engineers simply subtract the weight of the moisture from the total steam coming to the engine, and call the result the dry steam supplied. This is not a scientific procedure, but has certain practical points in its favor. For one thing, so far as the performance of the engine is concerned, the harm done by the entrained water in augmenting cylinder losses will far exceed any possible gain from its heat content. Under the quite common proviso that "commercially dry steam" is to be supplied, it may be said that if moisture is not removed by an efficient separator, the very least concession is not to charge the engine with any heat which it may carry. On the whole, however, it is better to go to the heat-unit basis, emphasizing thermodynamic efficiency rather than mere steam rate, and perhaps giving more commercial importance to relative efficiency. Above all, the character of any change from actual to equivalent or adjusted quantity should be clearly stated in the report of a test, especially if it is a departure from common practice, or if more than one usage is generally accepted.

§ 27. Examples of Performance

(*a*) SCOPE OF PRESENTATION. — The examples now to be set forth, together with those already given in § 22, cover the whole range of steam-engine practice, as will appear from a glance at the columns for size, speed, limiting pressures, etc., in Table 13. For graphical illustration, the simple pressure-volume diagram is chosen, in preference to

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any derivative from it, such as the temperature-entropy diagram. In every case, the direct indicator diagrams are replotted so that the performance of one pound of steam is represented, according to the scheme of § 21 (*l*). In Table 13, numerical values of data and of calculated results are arranged in systematic fashion, and from its columns a clear idea of the magnitude of the various quantities can readily be gotten. All of the diagram examples are included in the table, together with some others that have not been thus illustrated.

(*b*) SCHEME OF THE STEAM DIAGRAMS. — In regard to the combined or transformed diagrams in Figs. 129 to 140, the following points are to be noted:

All are laid out to the same scale of pressures and, with one or two exceptions, to the same scale of volumes: these have been chosen with regard to the space available on the page, and are used also for the simple-engine diagrams in Figs. 90, 92, 98, 99, and 109. Adherence to these established scales leads to marked distortion of form in some extreme cases, such as Figs. 133 and 134; but this is more than balanced, for present purposes, by the advantage of ready comparability.

Without exception, the scheme of combination is that of the full-line diagrams in Fig. 80, the clearance lines or axes of zero volume being brought to a common position.

Where the clearance effects are large, in Figs. 134 and 135, hyperbolas touching the expansion and compression curves of the respective diagrams are carried down or up to horizontal, intermediate reference lines, for a graphical comparison of the volume measures of the working steam — compare the slightly different scheme in Fig. 80. In every case, relative values of the net *pv* product, for selected points, are given under the letter M (standing for "measure"). From the total value of *pv*, at a point on the expansion curve, is subtracted the value for the clearance steam, from a point on the compression curve. The net *pv* at high-pressure cut-off or for the high-pressure cylinder is taken as unity, and the other values are expressed in terms of this unit.

Under the letter D (for "division") are given sets of ratios or fractions which show how the work per cycle or the power developed is divided among the cylinders or stages.

Regularly, the diagram is drawn for one pound of working steam, fed to the cylinder, or for more than one pound of total steam consumed in a jacketed engine. Exceptions, in which the jacket steam is included within the pound, are found in Fig. 135, where the jacket water was not measured, in Fig. 136, where there is a special purpose of comparing performance without and with jackets, and in Fig. 140, where the jacket

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water could not have been drained off without deranging the cycle. The reference curve drawn outside the expansion curves is the saturation line, for one pound of steam, laid out from column 2 of Table II; and the extra curve, of which just the beginning is dotted in on Figs. 129, 130, and 131, shows what would be the added volume of the steam condensed in the jackets.

Since the expansion curve represents the volume of all the steam in the cylinder, including the clearance steam, the missing quantity is not given directly by the distance between that curve and the saturation

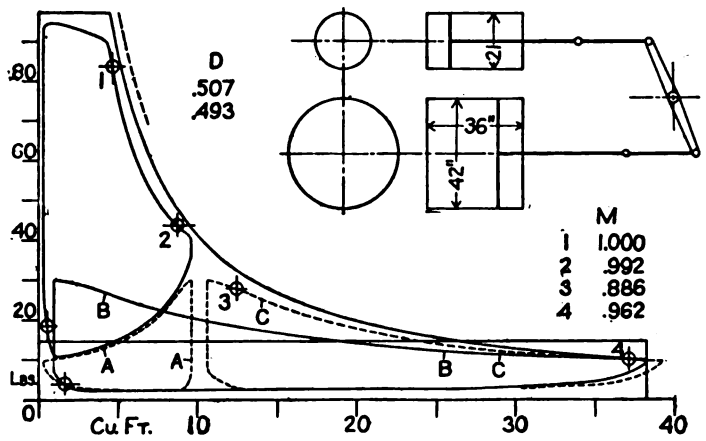

FIG. 129. — Direct-expansion Engine with No Receiver. Holly duplex compound pumping engine, 21 and 42 by 36 in., ratio 4.04, r.p.m. 20, full jackets. Engine at South Bethlehem, Pa., test by author in 1902. No. 34 in Table 13.

line: but its volume is shown, very approximately, by the difference between the intercepted width of the diagram and the abscissa of the one-pound curve. The comparison would be made more striking by transforming the diagrams so as to bring their compression curves to the line of zero volume, as in the dotted-line case of Fig. 80 and in Fig. 81; but the extra labor and the added confusion of lines on the reduced illustrations are good reasons for omitting this step.

On the single diagrams in Fig. 133, the saturation curves SS are drawn for the whole weight of steam expanding in the cylinder and the missing quantity is directly shown. This is feasible where there is but one cylinder, so that the quantity of clearance steam does not change, and where several diagrams are not referred to the same axes, as is done in Figs. 134, 136, etc.

Each diagram here given is at least the mean of a full set of indicator cards; with a duplex engine, as in Figs. 129, 133, 134 II, and 137, this

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signifies that four sets of ordinates have been measured and averaged. In some cases the combined diagrams are the mean of a number of sets of indicator diagrams, representing the whole of a long test.

The essential facts about the engine are given beneath the figures. Other information, with derived results as to performance, will be found in Table 13.

(c) ENGINES WITH SMALL COMPRESSION. — In Figs. 129, 130, and 131 are given diagrams which show a minimum of departure or loss from the simple, "ideal" form of Fig. 57 and Fig. 76. They are all from pumping engines, necessarily slow-running: the low speed makes the kinetic losses small, while permitting small ports and low clearance ratios; at the same time, its possible harmful effect in the direction of cylinder losses is neutralized by the influence of steam jackets and reheating receivers. The diagrams are well filled out, or the diagram factor, as defined in § 29 (b), is high.

Figure 129 does not represent especially good performance, but is chosen to illustrate the type of compound engine with absolutely no receiver. The two pistons are coupled together through a vertical rocker beam, as outlined above the diagram; the strokes are simultaneous and opposite, and a single valve between the cylinders (at each end) controls high-pressure exhaust and low-pressure admission. The low-pressure diagram is drawn in three different ways: first, at A, it is placed beneath the high-pressure diagram, on the same base line (compare Fig. 77), chiefly in order to show how nearly the two lines of common expansion agree; next, at B, it is stretched out to the full length representing the volume of the low-pressure cylinder, as in all the other figures; finally, at C, the length is made equal to the increase of volume during the common expansion. At the beginning of the stroke, the communicating volume comprises the high-pressure cylinder V_1 and the two clearances; diagram C begins at this distance from the vertical axis, and its length is $(V_2 - V_1)$. Evidently, this last scheme gives the best test of the continuity of expansion. As regards the representation of work, it will be noted that areas A and C are together equal to area B; the simplest interpretation of the diagrams is to think of the low-pressure piston area A_2 as divided into two portions, A_1 the same as the small piston, and $(A_2 - A_1)$. The former receives the work of diagram A, the latter that of C. Only in the particular case of no expansion in the low-pressure cylinder alone is there need or use for this expedient.

Figure 130 shows a performance of higher excellence. The combined effect of low speed and full jacketing is seen in the unusually marked reëvaporation toward the end of the high-pressure expansion, and in the increase of the pv measure; reference to Table 13 will show an

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unusual increase in the indicated steam from initial cut-off to final release. The mechanism connecting the two pistons is essentially the same as in the preceding engine, although different in its arrangement

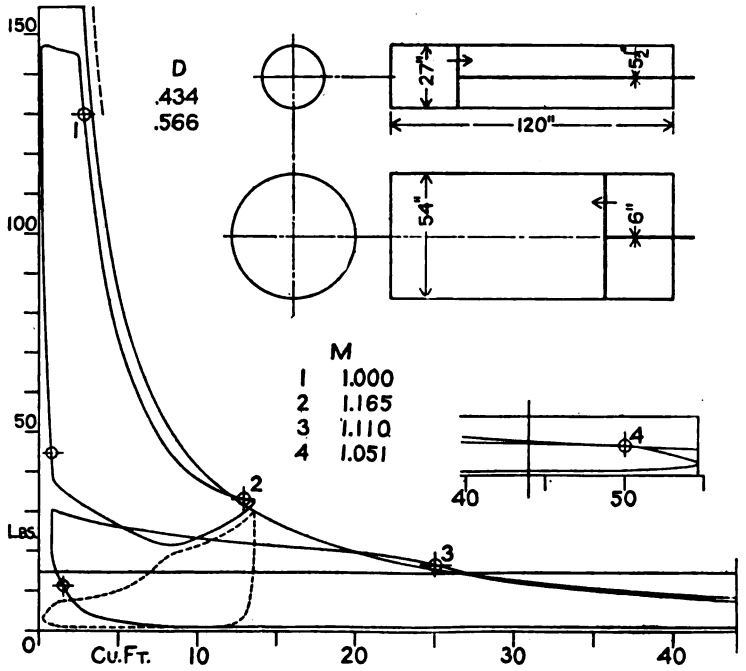

FIG. 130. — Direct-expansion Engine with a Receiver. Leavitt vertical compound beam-fly-wheel pumping engine at Louisville, 27 and 54 by 120 in., ratio 4.02, r.p.m. 18.6, full jackets and reheater. F. W. Dean, 1894, Trans. A. S. M. E., Vol. 16, page 169. No. 35 in Table 13.

and proportions, and there is a receiver of considerable size between the cylinders.

The standard type of pumping engine for large water works is now the vertical triple-expansion. From the highest development of the compound, in Fig. 130, to an early example of the triple, in Fig. 131 I, is a very short step. Between I and II the chief difference is an increase of steam pressure from about 120 to 150 lb. by gage. The excellence of these diagrams is patent, and they call for no comment. To supplement the data in Table 13, extending them to performance in the pump cylinder, the results in the tabulation on the next page are added. The water horse-power is calculated from the effective pressure pumped against, which is measured by gages, as explained in § 26 (*j*), so that the mechanical efficiency E_m includes the effect of some

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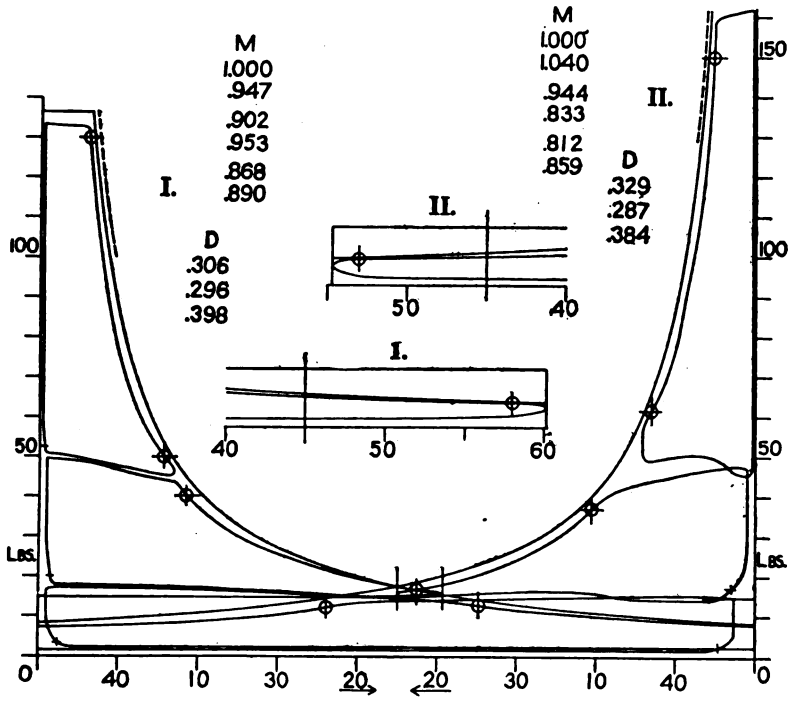

FIG. 131. — Two Vertical Triple-expansion Pumping Engines, with jackets and reheaters.

I. Allis-Chalmers Engine at Milwaukee, 28, 48, and 74 by 60 in., ratio 7.11, r.p.m. 20.3 R. C. Carpenter, 1893, Trans. A. S. M. E., Vol. 15, page 313, also *Engineering Record*, Dec. 2, 1893, No. 36 in Table 13.

II. Snow Engine at Indianapolis, 29, 52, and 80 by 60 in., ratio 7.66, r.p.m. 21.2. W. F. M. Goss, 1898, Trans. A. S. M. E., Vol. 21, page 793. No. 37 in Table 13.

RESULTS FROM PUMPING ENGINES.

No. in Table 13.	Figure.	Steam horse-power.	Water horse-power.	Mech. efficiency E_m	Steam per hour,		Duty per 1000 lb. steam.
					Steam horse-power.	Water horse-power.	
34	129	281	246	0.921	15.63	16.97	116,700,000
35	130	643	599	0.931	12.22	13.22	149,700,000
36	131 I	574	518	0.902	11.80	13.07	151,100,000
37	131 II	783	735	0.938	11.50	12.26	161,600,000
(a)	.....	*866*	823	*0.95*	*10.31*	10.86	182,400,000
38	.....	926	879	0.95	9.65	10.15	195,000,000

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hydraulic friction. The duty here given is per thousand pounds of steam; to get that per million B.t.u., apply E_m to the absolute thermal efficiency E in Table 13, then reverse the calculation shown in Eq. (139). On this latter basis, 165 million at the plungers, or 175 million at the steam pistons, is about the limit of performance of the engine working on the Rankine cycle.

The test marked (a), not given in Table 13, is from a Holly engine at Philadelphia, 30, 60, and 90 by 66 in., at 20.1 r.p.m., with steam at 180 lb. by gage, carrying 0.0114 of moisture: the values in italics are based on an assumed 0.95 efficiency. Test of March, 1910, from Bulletin of Holly Manufacturing Co. In test 38, the steam is superheated.

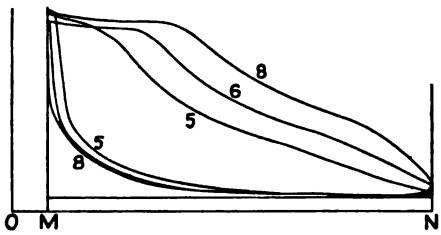

FIG. 132. — Mean Indicator Diagrams (four cylinder ends), from simple locomotive, 21 by 30 in., No. 2 in Pennsylvania Railroad Tests at St. Louis Exposition, as listed under Fig. 103.

(*d*) ENGINES WITH LARGE COMPRESSION. — This class is in strongest contrast with that just considered. The large clearance which is implied results partly from using wide ports, in adaptation to high speed, partly from the use of a single valve, requiring long ports; while long compression is an unavoidable accompaniment of early cut-off, with a single valve, as is shown by the typical diagrams in Fig. 132 — see also § 39. The class embraces the small, high-speed, automatic cut-off engine, represented by Figs. 2, 5, and 7, the locomotive, and the marine engine.

The general type of diagram from the stationary, high-speed engine is sufficiently illustrated in Figs. 84 and 78. Several diagrams from a simple locomotive, the engine with the greatest range of pressure in a single cylinder, are given in Fig. 132 and are transformed to our uniform system in Fig. 133. Thus replotted, they show clearly in what part of the whole range of expansion the locomotive has its field of operation, and how well that field is covered. To test the conformity of the expansion curve to the hyperbola $pv = C$, a short piece of the latter, as produced from cut-off at C, is drawn outside the release R in tests 6 and 8; in test 5 it falls right on the expansion curve at R.

The performance of the compound locomotive is represented by

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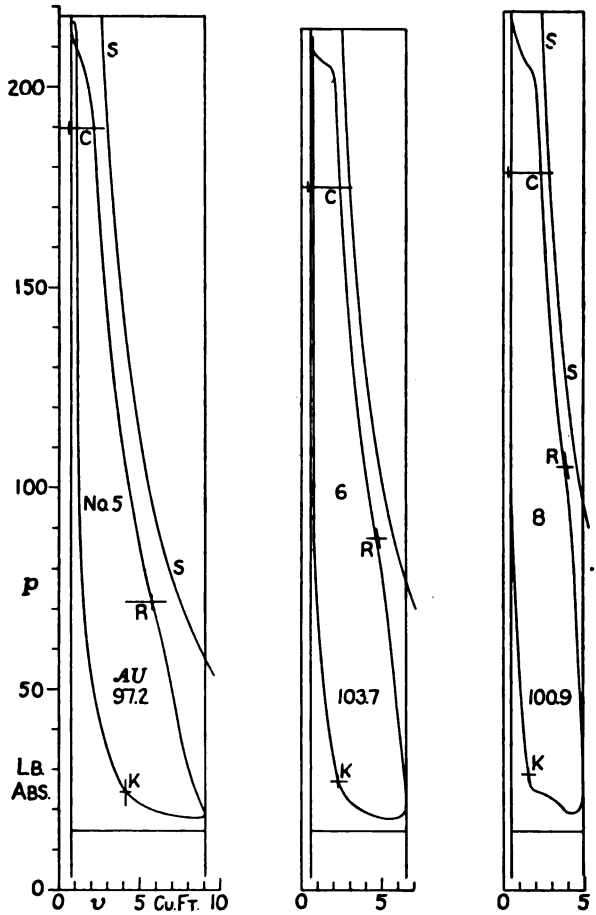

FIG. 133. — Diagrams in Fig. 132, on "unit" system, with same scales as other figures in this Section. Volumes for one pound of working steam, but steam curves SS for total steam expanding, including clearance steam. Weights as follows:

Test.	Cut-off.	Steam weight.
5	0.173	1.250
6	0.307	1.147
8	0.407	1.110

These three tests are averaged together as No. 6 in Table 13.

Fig. 134. The clearance effects are large, especially in case B of figure I, where the high-pressure diagram is shoved out more than half-way beyond the one-pound saturation curve. We have here a striking example

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of the influence of speed upon steam distribution; as noted beneath the diagrams, the cut-offs (by the valve) are nearly the same in cases A and B; but at the higher speed the pressure at cut-off is so much lowered by valve and port throttling, and the pressure at compression is so much

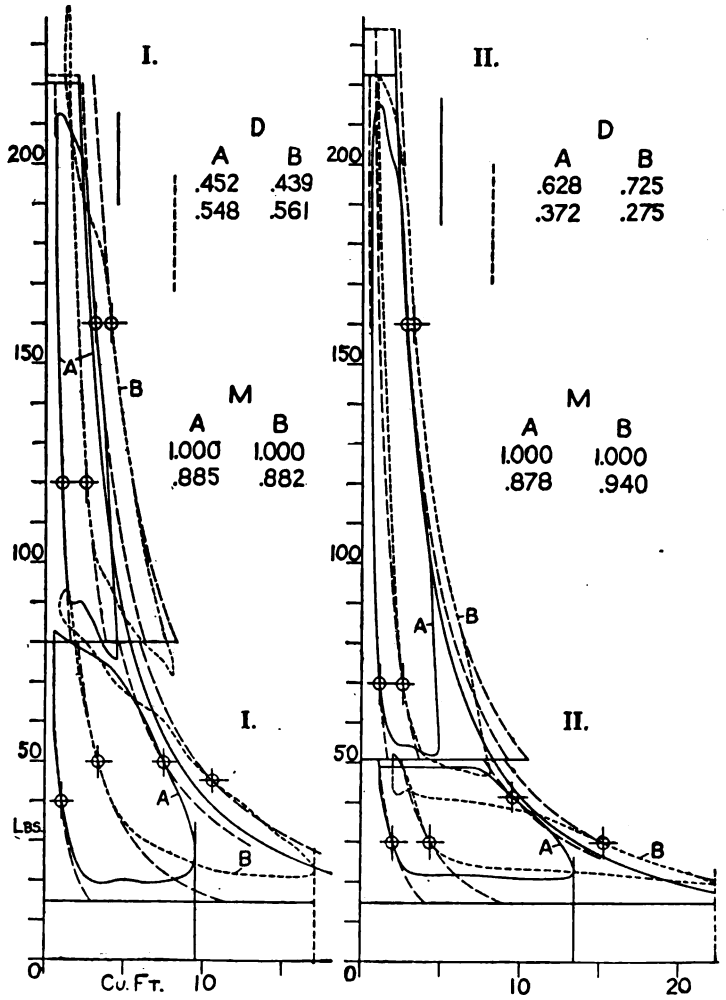

FIG. 134. — Diagrams from Compound Locomotives, Pennsylvania Railroad Tests, St. Louis, 1904.

I. Cross-compound (two-cylinder), Consolidation type, 23 and 35 by 32 in., ratio 2.34: No. 3 under Fig. 103, No. 7 in Table 13. Diagram A, test 308 (in original report), r.p.m. 80, cut-off 0.53; B, test 312, r.p.m. 160, cut-off 0.50.

II. Four-cylinder, Atlantic type, 14.2 and 23.7 by 25.2 in., ratio 2.81: No. 5 under Fig. 103, No. 8 in Table 13. Diagram A, test 508, r.p.m. 160, cut-off 0.50; B, test 512, r.p.m. 240, cut-off 0.34.

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raised, that the relative amount of clearance steam is more than doubled. In II the receiver pressure is abnormally low, with the result of ex-

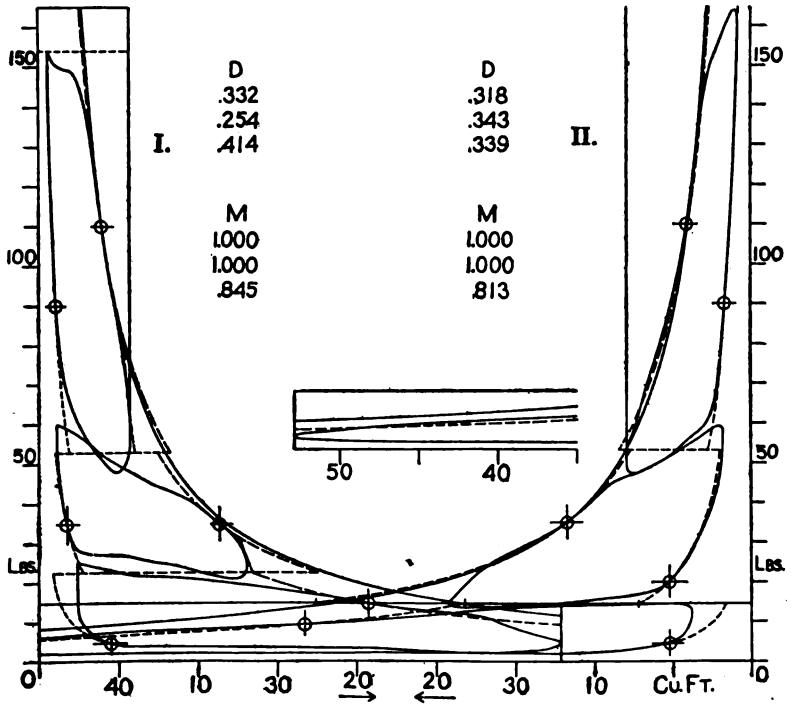

FIG. 135. — Triple-expansion Marine Engines, Tests by Committee of the Institution of Mechanical Engineers.

I. Steamship *Meteor*: 29, 44, and 70 by 48 in., ratio 5.70, r.p.m. 72, jackets on all cylinders. Proc. I. M. E., 1889, and *Engineering*, 1889 I, page 527. No. 11 in Table 13.

II. Steamship *Iona:* 22, 34, and 57 in. by 39 in., ratio 6.75, r.p.m. 61, jacket on high-pressure cylinder only.] Proc. I. M. E. 1891, and *Engineering*, 1891 I, page 568. No. 12 in Table 13.

cessive drop loss in case A. Besides large clearance effects, the characteristics of these diagrams are large kinetic losses and high back pressure — the last due to the exhaust nozzle, and an inherent element in the operation of the locomotive. Locomotive performance is represented by tests 6 to 10 in Table 13: in spite of the losses just named and of the strong cylinder action shown in Fig. 105, quite a fair relative efficiency (compared with the Rankine cycle) is realized.

The diagrams in Fig. 135 are typical of the marine engine, although they are from rather small ships, and with not very high steam pressures: in some lines of service, pressures of 250 lb. by gage are common. Comparatively few steam-consumption tests of marine engines have

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been made: the difficulties are greater than on land, and the simple coal test of the whole plant is generally considered enough. Diagram II shows an unusually long expansion: total contained volumes of 25 to 35 cu. ft. per pound of steam (including clearance) are usual, as against 50 cu. ft. and more in Figs. 130 and 131. This short expansion and the large kinetic losses and compression effects account for the comparatively high steam rates of marine engines. In tests 11, 12, and 13 of Table 13, S_h ranges from 13 to 15 lb. In warship engines more closely designed as to size of cylinders, 15 to 16 lb. is the common range of steam consumption — see data collected in Table 75D of STEAM ENGINE, Vol. II. In the poor relative efficiency of the marine engine is seen the reason why the turbine surpasses it in economy: as against good stationary engines, the turbine has no great advantage in thermodynamic performance, except in the better utilization of vacuum.

(*e*) COMPARISONS OF STEAM ACTION. — We now take up power generating engines of the Corliss class — this name of the most prominent member being given to the whole class of engines which have the same general kind of steam distribution: they are characterized by small clearance and compression, but in the matter of kinetic losses lie between the low-speed pumping engine and the slide-valve type. The first two examples, brought together in Fig. 136, present a most instructive comparison between the systems of nonuse and use of steam jackets — each engine being properly designed for its particular conditions. In A we have a large engine at fairly high speed, the combination of size and speed being such that jacketing would have little, if any, useful effect: the engine had a reheater in the receiver, but even that was left out of action during the tests. In B, on the other hand, the engine is smaller, the speed is low, and the ratio of expansion is somewhat greater; jackets are fully applied, to barrels and heads of the cylinders, and to the receiver. Note that diagram B is drawn for one pound of total steam consumed, the jacket steam being included under the one-pound saturation curve. For the cylinder feed alone, the constant-weight curve (of which the beginning is dotted in) would be laid out for 0.881 lb., since the jackets use 0.119 of the steam supplied.

Above the steam diagrams are given quality diagrams for the two cylinders, showing the ratio of indicated to actual steam weight in the cylinder during expansion: these are on a stroke-line base, instead of the pressure base in Fig. 82 and in Fig. 138. The relative heights of the high (H) and low (L) quality curves in the respective cases are typical: the unjacketed engine shows a decrease in the steam fraction x from first to second cylinder, the jacketed engine an increase; and

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the resulting increase in the work of the low-pressure stage is the principal gain from jacketing. The different ways in which the product pv changes (as indicated under M) are also typical: it falls off much more rapidly without the jacket.

Between these two engines there is a balancing of gains and losses which leads to almost an equality in thermal efficiency — see Table 13. Engine A has slightly superheated steam and realizes a much

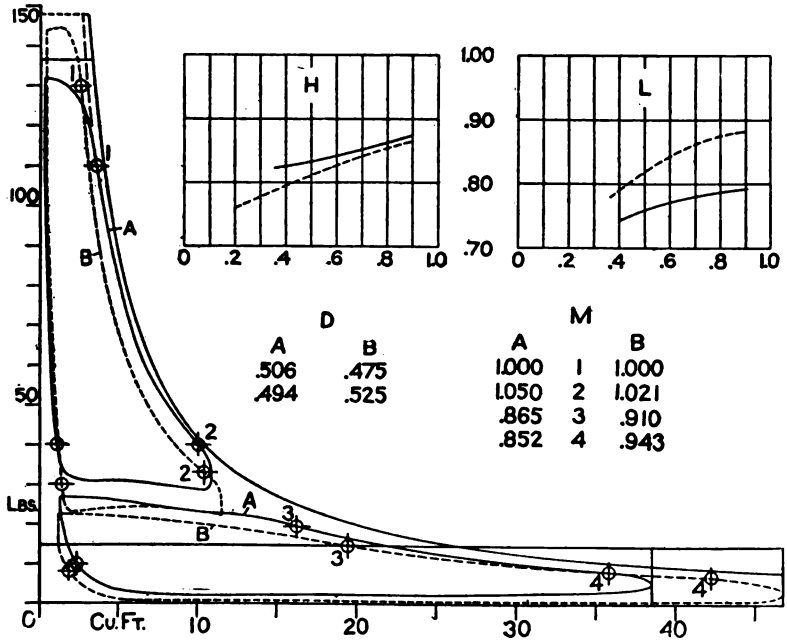

FIG. 136. — Comparison of Corliss Engines Without and With Jackets.

A. Horizontal, cross-compound, plain receiver, no jackets, cylinders 30 and 56 by 72 in., ratio 3.48, r.p.m. 65. J. E. Denton and others, 1893, Trans. A. S. M. E., Vol. 15, Page 882. No. 24 in Table 13.

B. Compound beam engine, reheater, full jackets, cylinders 17 and 34 by 60 in., ratio 4.09, r.p.m. 34. M. Longridge, 1895, *Engineering*, 1895 I, page 132. No. 25 in Table 13.

fuller steam volume, especially in the high-pressure cylinder; B has the advantage of higher initial pressure and longer expansion, and shows smaller kinetic losses between the stages. If the two were brought to common limits of action (initial pressure and final volume), the situation would be summed up by saying that in engine B the special heating devices do not quite succeed in overcoming the handicap of smaller size and lower speed.

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The example in Fig. 137 has no direct bearing on the preceding comparison, except that, in a very large engine at good speed, it shows high steam quality without the use of any auxiliary heating devices. The principal interest lies in the change from ordinary condensing

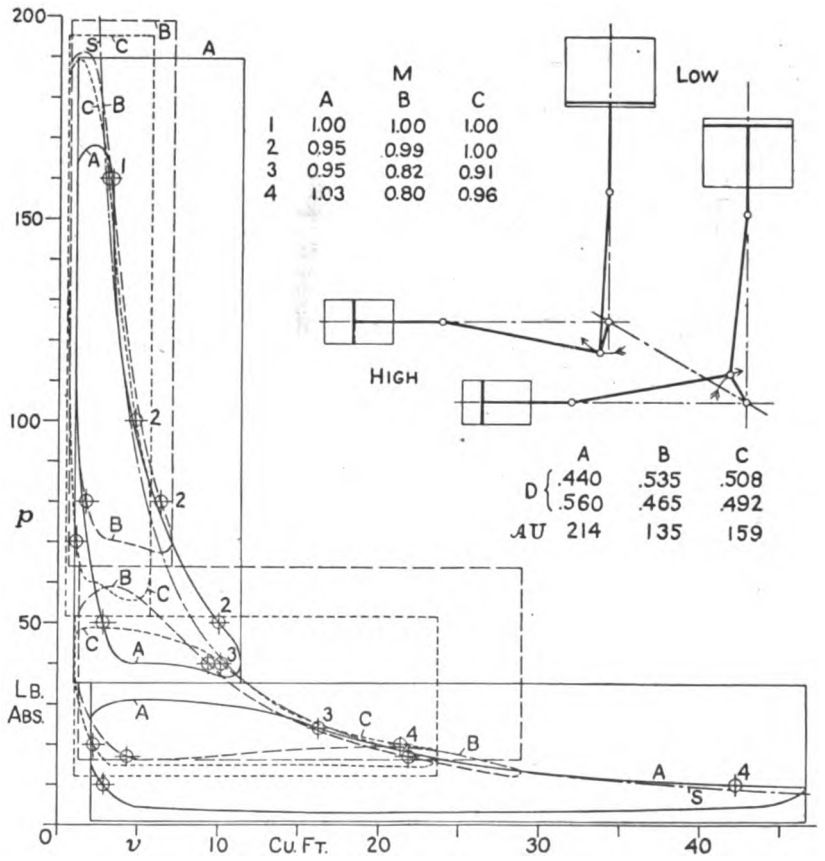

FIG. 137. — Large Corliss Engine Driving Electric Generator. Duplex, cylinders 42 and 86 by 60 in., ratio 4.30, r.p.m. 75, neither jackets nor reheater. Interborough Power House, New York City: H. G. Stott, Jour. A. S. M. E., Mar. 1910, Vol. 32, pages 315 to 372. No. 27 in Table 13.

operation, test A, to what is essentially noncondensing action, tests B and C; this change permitting a low-pressure turbine to be placed in series with the engine. Reference to Table 13 will make it evident that we have here a striking illustration of the matter discussed in § 26 (*k*). The relative efficiency E_R is only 0.60 with vacuum exhaust; but when the exhaust pressure is raised the engine is able to realize from

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0.75 to 0.80 of the ideally possible performance, in spite of the large kinetic losses so apparent in the combined diagrams. Smallness of the missing quantity, together with a fairly large proportion of clearance steam, causes the high-pressure expansion curves to lie well outside of the one-pound volume curve, and brings those of the low-pressure cylinder quite close to that curve. Along with work-division D is given the work AU per pound of steam, found from the diagram by planimeter and expressed in B.t.u.: that these values do not quite agree with those under W in Table 13 is due in part to unavoidable inaccuracy in redrawing Fig. 137 from published diagrams, and perhaps also to the original use of indicator cards which were not quite the average for the run.

(*f*) Effect of Jackets and Reheaters. — In regard to the economical effect of these devices, examples can be found which range all

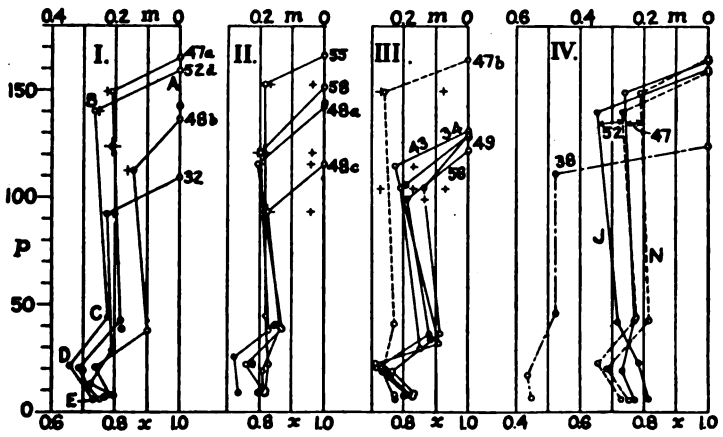

Fig. 138. — Quality at Critical Points, Compound Engines of Corliss or equivalent types, Barrus' Engine Tests: original numbers used to designate tests.

Group I. No jackets or reheaters.
Group II. Reheaters only.
Group III. Jackets and reheaters: No. 47 not included in means.
Group IV. Same engines, with and without jackets and reheater (full and dotted lines respectively). No. 38 is added as an example of excessive leakage.

the way from 20 per cent gain to nearly 10 per cent loss, using steam per horse-power-hour as the basis of comparison; but the omission of exceptional cases will cut these limits down to about plus 10 and minus 3 per cent. The only such pair of tests in Table 13, No. 29, shows a slightly greater consumption with the jackets, which is just balanced by the gain in feed temperature. Partly as further information along this line, partly to show how erratic are the data available, a number of results from engines working under service conditions are diagrammed in Fig. 138, on the scheme of Fig. 105. The " quality " shown is the

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ratio of indicated steam to actual steam consumed: since clearance steam is not included, the fraction represented is not quite the true quality of the steam in the cylinder; further, the jacket steam here forms a part of the "missing quantity." In the first three groups, average points are found and marked by blacked circles; also, results from Eq. (126), for the missing steam at initial cut-off, are indicated by + marks: the values of m_f are laid off directly in group I, but in II and III they are measured (after proper reduction) to the left of first + marks which show the proportion of steam used in jackets and heater — except that in two tests of group III the latter quantity was not found. These comparisons have interest in connection with the last columns on page C of Table 13.

Comparing means in groups I to III, we note practically the same realized volume at the initial cut-off in all three cases, but decidedly stronger reëvaporation in the first cylinder with the jackets in action. From I to II there is a good gain in quality in the lower stage, hardly maintained in III. Group IV contains two pairs of comparative tests, each pair from the same engine under different conditions, and agrees with Fig. 136 in showing better quality in the high cylinder without the jackets, in the low cylinder with them. Test 38 is put in merely to show how greatly the missing quantity may be increased by leakage.

Experience in the use of these heating devices can be summed up in the simple statement that with large engines, at good speeds and well loaded, their effect is nearly neutral, although generally inclined to the side of a small gain. As the conditions of operation are more conducive to cylinder losses, jackets are more beneficial: thus a jacket system which produces very little net benefit at full load, and causes a loss at overload, will quite strongly promote economy under a light load, with its excessive ratios of expansion. The old *Michigan* test, Fig. 92 and No. 18 in Table 13, is a case where very decided improvement would have resulted from the use of a steam jacket.

The function of the reheater is to furnish dry, and preferably superheated steam to the low-pressure cylinder. A thoroughly sound view of the principles involved is embodied in the scheme, applied in some of the best-designed engines, of passing the exhaust from the higher cylinder through a regular "steam separator," thus removing most of the contained water mechanically, and then sending it over the heating surface. The expenditure of live steam to evaporate water into steam which can at best work through but a part of the whole temperature range, is not a logical and effective device for increasing economy.

(*g*) EFFECT OF SUPERHEATING. — A small or poorly placed superheater, or a long and poorly insulated steam pipe, may give only the

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slight superheats of 10 to 15 deg. seen in several of the tests in Table 13: this amounts to little more than furnishing dry steam and throwing in a little extra heat for good measure. With the superheater incorporated into the construction and into the hot-gas circulation of the boiler, there is a moderate range, say from 50 to 150 deg. above saturation. Very high superheats, approaching the limit of about 400 deg., and involving steam temperatures as great as 750 deg., are usually and most safely produced in separately-fired superheaters. To avoid overheating of the tubes, the supply of heat to their surface must be more closely and carefully regulated when it is being abstracted by a current of hot steam than when a body of relatively and definitely cooler water is present; such regulation is more difficult when the superheater is a part of the main boiler, and if it is placed too near the fire there is serious danger of its being " burned out " when the boiler is forced.

The proper basis of judgment upon the thermal economy of superheating is found, not in the steam rate, but in the thermal efficiencies, both absolute and relative. Referring to Figs. 55 and 56, we readily see that if the engine can maintain the relative efficiency E_R undiminished as superheat is added, there will be some gain in absolute E, because the ideal performance is better when a part of the heat is received at temperatures above that of saturation; if E_R increases, yet more advantage will be gained. Consider the example diagrammed in Fig. 139 and entered as No. 30 in Table 13. Comparing superheated steam at 737 deg. with saturated steam at 362 deg., we see steam consumptions in the ratio of 100 to 145: in the thermal quantities, letting 100 represent the value with common steam, the relative numbers with superheated steam are:

Heat supplied per pound of steam	119
Output of Rankine cycle	129
Actual output	145
Absolute efficiency	122
Relative efficiency	112

These do not quite correspond with the results in Table 13, because they are got by referring both tests to an exhaust pressure of 2 lb. abs., according to the scheme of Table 12, page 242. The most important showing is the 22 per cent gain in absolute efficiency. This is unusually high for an engine of the naturally economical type, gains of 6 to 12 per cent being more common. The greatest improvement has been found in certain small, simple engines. As an instance may be quoted some tests made by Professor Ripper and reported in Proc. Inst. C. E., 1896, Vol. 128, page 60: the engine was a 7 by 12 in. Schmidt

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superheat "motor," run noncondensing at 180 r.p.m.: with steam at 130 lb. abs. and superheated about 330 deg., the consumption per horse-power-hour was 17 to 18 lb., while with saturated steam, dropped to 100 lb. abs., the consumption rose to 38 lb. There is strong reason to believe that the saving of steam in such engines is in large measure due to a diminution of leakage — compare the valve leakage described in § 22 (*p*), which is of a kind that would be greatly reduced by superheating.

Against the large saving in steam consumption, typically represented by tests 10 and 30 to 33 in Table 13, and the smaller yet considerable gain in thermal efficiency, certain disadvantages must be charged. One is the more rapid deterioration of the superheater, as compared with the boiler, because of excessive temperature of the metal. This causes trouble in piping and in engine also, as affecting the material to some degree, and as making necessary greater allowance and freedom for expansion and contraction. Lubrication of the valves and piston is more difficult with superheated steam: at one time in the history of the engine, this was a principal influence leading to the abandonment of superheating; when it was revived, beginning in the early 1890's, better lubricants were available, in the way of high-grade mineral oils. Now the cylinder lubrication of a steam engine with high superheat is certainly no more difficult than that of the large gas engine.

(*h*) PERFORMANCE WITH HIGH SUPERHEAT. — This is well exemplified by Fig. 139, where comparative tests from the same engine are laid out on the same one-pound basis. A separately-fired Schmidt superheater produced the high temperature noted; as already remarked, this type of apparatus has the advantage of controllability, but the economic disadvantage that the standby loss, or the amount of coal that must be used for banking the fire overnight, will probably be relatively greater than in the larger furnace of the boiler. A portion of the main steam supply was shunted through the reheater, then turned back into the current. The engine had lift or poppet valves in the high-pressure cylinder, Corliss valves in the low. In spite of the cooling effect of the reheater coil and of the cylinder walls, the steam is quite strongly superheated at high-pressure cut-off, and somewhat superheated at release; and after reheating it is still above saturation during the first part of the low-pressure expansion. That the heat interchanges are small is shown by the close approach of the expansion curve to the adiabatic form: thus if the coördinates of points 1 and 2 of diagram A are substituted in the equation $p_1v_1^n = p_2v_2^n$, the value of n is found to be 1.26 for a curve which will pass through these

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points; while for a true adiabatic n would be a little over 1.3 — see § 14 (*e*).

This marked change in the form of expansion exerts quite an influence upon the division of work between the cylinders, as is shown both by the numbers under D and by the indicator diagrams, which represent practically the same development of power. The specific

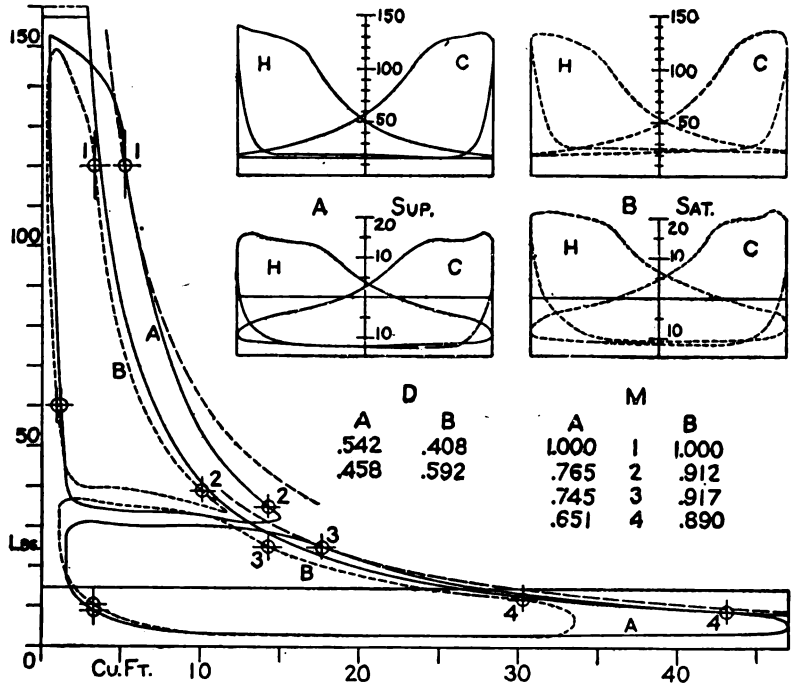

Fig. 139. — Compound Engine with Highly Superheated Steam. Horizontal, cross-compound, reheater, no jackets; 16 and 28 by 42 in., ratio 3.08, r.p.m. 102; test A with steam superheated 375 deg., test B with saturated steam. D. S. Jacobus, 1903, Trans. A. S. M. E., Vol. 25, page 264. No. 30 in Table 13.

steam volumes are relatively so much smaller in the lower stage when working from high superheat that the cylinder ratio is less for the same completeness of expansion (as measured by the pressure at release). The three to one ratio in this engine is about equivalent in effect to a four to one ratio with saturated steam.

(*i*) The Regenerative Cycle. — An actual example of the scheme outlined in Fig. 59 is given in Fig. 140. The feed water, drawn from a hot-well tank, first passes through a surface heater in the exhaust line, between the low-pressure cylinder and the condenser; it then goes through a series of four mixing heaters, being pumped from each lower one into the next, in which there is a higher pressure and temperature.

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Steam for these heaters is taken from the low-pressure cylinder at release (intermittently, through a special valve), and from the third, the second, and the first receivers — the jacket and reheater drains

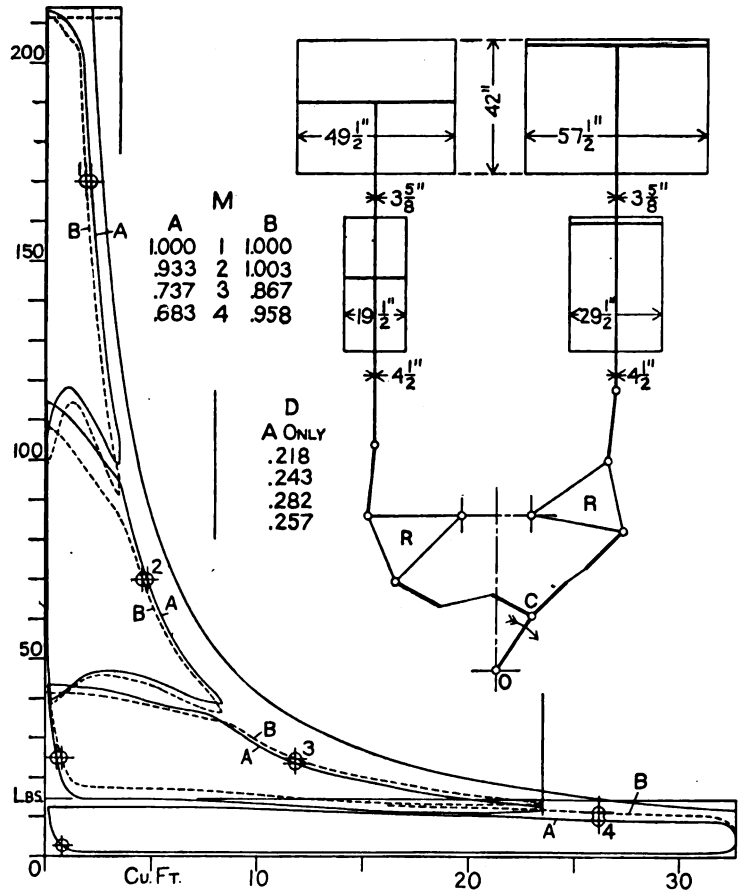

FIG. 140. — Engine with Regenerative Feed Heaters. Nordberg quadruple engine at Pittsburg; cylinder sizes given on diagram, ratio 9.08, r.p.m. 36.5, jackets and reheaters. R. C. Carpenter, 1898, Eng. Record, Apr. 22, 1899; also R. H. Thurston, Trans. A. S. M. E., Vol. 21, page 203. No. 39 in Table 13. Compare Fig. 141, page 273.

being included in the steam and water thus abstracted from the working cycle. The last heater delivered water at 311 deg., as compared with a steam temperature of 388 deg. in the boiler; while with the similar engine listed as No. 40 in Table 13, the feed temperature was 334.5 deg. and that of the steam 403.5 deg.

The normal working of the engine is represented by the full-line

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diagrams in Fig. 140. The relative decrease in effective volume as the pressure falls, due to the diminution in the quantity of steam passing through the cylinders, is clearly shown, graphically by comparison with the dotted diagrams, numerically by the ratios under M for case A. The main plot is made for one pound of total steam consumed, since that used in the jackets could not well be determined, except by inference from thermal changes. The diagrams in dotted line are laid out on the same base line, or for the same power, hence do not represent the working of one pound of steam; they show performance without the special feed heaters, but with the jackets still in action.

The ideal output and the relative efficiency for engines of this class, Nos. 39 and 40 in Table 13, are based on the Carnot cycle efficiency, as in Eqs. (96) and (97). Decidedly, steam per horse-power-hour is not the proper criterion of performance when comparing tests like Nos. 38.2 and 39, where there is a difference of 25 per cent in S_h but practical equality in thermal efficiency. Any jacketed engine in which the jacket water is either returned to the boiler at full pressure or mixed with the feed water so as to raise the latter above the exhaust temperature goes a little way from the Rankine cycle toward the regenerative, form, but not far enough to call for any change toward the Carnot basis of comparison.

(*j*) Scheme and Showing of Table 13. — The various classes of engines represented are indicated by the division titles on page B. The table includes all the engines of which steam diagrams have been given in this section, many of those represented by the steam-consumption diagrams in § 22, and others selected to fill out the illustration of typical performance. It must be borne in mind that the results here shown are, in most cases, somewhat better than the average working of engines of the several classes. Special tests, of sufficient importance to be fully published, are generally made when the engine is comparatively new and in good condition, with a minimum of leakage; and there is a natural tendency for poor results to be suppressed rather than to get into print.

The notation of the table will now be described and defined, with some remarks as to the range of variation of the more important quantities. To facilitate comparison with reports in the metric-centigrade system, the reduction ratios of the differing units are given beneath the definitions.

When comparing these engine-test results with those from turbines, as given in Table 20, it must be borne in mind that here indicated power is the measure of output, there shaft or brake power. For machine efficiency of the engine, see § 28, following.

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PAGE A. CONDITION OF OPERATION, SIZES OF CYLINDERS, SPEED AND EXPANSION DATA.

No. — A serial number is given to each engine; decimal figures designate different tests of the same engine.

CONDITION. — By a condensed notation a number of important general conditions of the test are stated, as follows:

First letter — condition of steam:

T = saturated steam;

P = superheated steam.

Second letter — kind of exhaust:

A = atmospheric exhaust, or noncondensing;

C = condenser or vacuum exhaust.

Numeral — number of stages in the expansion.

Third and fourth letters — condition as to jackets and heaters:

J = jackets in use;

H = reheater in receiver in use;

N = no jackets or heaters.

If either J or H is given alone, absence of the other type of heating device is implied.

Last letter — manner of measuring steam consumption:

F = feed water weighed or measured;

M = feed water metered;

C = steam condensed and weighed.

N Speed in revolutions or double strokes per minute.

V Piston speed in feet per minute — see § 3 (*d*).

(1 meter per sec. = 196.7 ft. per min.)

e Apparent cut-off in high-pressure cylinder, or fraction of stroke completed at cut-off, not counting the clearance.

r Ratio of expansion, or ratio of the final full volume of the low-pressure cylinder to the volume back of the high-pressure piston at cut-off, clearance being taken into account. If R is the cylinder ratio and i_1 and i_2 the clearances as given above,

$$r = \frac{R\,(1 + i_2)}{e + i_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (140)$$

Here the cut-off ratio e is measured at a pressure below that of admission, so that the expansion ratio r does not show quite the full range from the effective initial volume (as at H, Fig. 73) to the final volume in the low-pressure cylinder. The latter showing is preferable, but in some cases the pressure at cut-off was not given.

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Diameters. Stroke. These, with the clearances given on page B, fix the essential dimensions of the engine as regards steam action.

Ratio. The Ratio (R in Eq. 140) lies between mean piston areas or piston displacements in the low-pressure and the high-pressure cylinders, passing over the intermediates in a triple or quadruple engine. Some discussion of cylinder ratios will be found in § 29 (*i*). Note the one case of two cylinders in a stage, in No. 33, which is a four-cylinder triple, having two low-pressure cylinders in "parallel."

Page B. Clearances, References, Etc.

Clearances. These are mean values for the first and last cylinders, expressed in the usual fashion as fractions of the displacement or nominal cylinder volume. Note the extreme range from 0.18 in locomotive No. 9 to less than 0.01 in pumping engines Nos. 36 and 39. These exceedingly low values are secured by using single-disc lift valves in the low-pressure cylinder, as against Corliss valves in the higher cylinders.

References. In many cases it will be necessary to go back to one of the steam diagrams to get a full reference. The principal sources, covering all but five of the American tests, are the book Engine Tests, by G. H. Barrus, the book of Locomotive Tests and Exhibits, by the Pennsylvania Railroad Company at the St. Louis Exposition of 1904, and the Transactions of the American Society of Mechanical Engineers. But a few European tests are included in this table. The more extensive tables in Steam Engine, Vol. II, cover a wider range.

Page C. Steam Conditions, Power, and Steam Consumption.

m Fraction of moisture in steam received by engine.

s Degrees fahrenheit of superheat, at engine when so given in original report, otherwise at superheater.

p_1 Absolute steam pressure, in pounds per square inch. This should be measured in the steam pipe, just above the shut-off valve at the engine; in some of the tests it was taken at the boiler only.

(1 kg. per sq. cm. = 14.22 lb. per sq. in.)

p_0 Absolute exhaust pressure, reduced to pounds per square inch, as explained in § 12 (*b*). Marked E, this is pressure in the cylinder, from the indicator diagram; marked C, it is in the condenser or the atmosphere: see also t_0, page 267.

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p_m Mean effective pressure, reduced to the low-pressure piston in all multiple-expansion engines — see § 21 (*g*).

H Indicated horse-power.

(1 metric h.p. = 0.9863 English h.p. — see § 26 (*a*).

S_h Steam per horse-power-hour, in pounds: this is the actual steam consumption, not corrected for moisture or superheat as described in § 26 (*n*). It includes all steam used in jackets or reheaters. Compare summary in § 26 (*b*).

(1 kg. per met. h.p. = 2.235 lb. per Eng. h.p.)

j The fraction of the total steam supplied to the engine that is used in the jackets and reheaters. A blank in this column indicates the absence of the quantity from the test.

S_{ic} Indicated steam at initial cut-off, in same terms as S_h, calculated by the methods in § 21 (*f*) and (*g*) — compare also the steam quantities described in § 21 (*k*).

S_{ir} Indicated steam at final release, similar in terms to S_{ic}.

m_c Fraction of the steam entering the cylinder, or of $S_h(1-j)$, which is not shown by the indicator diagram, representing the difference between this cylinder feed and the i.s.c. S_{ic}. Only where j is unknown and represented by a (?) does this "missing fraction" include the jacket steam.

m_f Missing-steam fraction calculated from the initial-condensation formula, Eq. (126), for comparison with actual m_c. In the large-compression group, tests 1 to 12, m_f does not come directly from the formula, but is increased in the ratio of total steam to working steam, as is done in § 22 (*k*), for Fig. 103.

Taking m_f as a standard of comparison, the following points may be noted, in brief supplement to and summary of the full presentation and discussion of data in § 22:

The formula is properly applicable to engines without jackets, and with not more than a beginning of superheating. In the small-compression class, engines 14 to 18, 20, 21, and 27 come under this definition; and with the exception of tests 15.1, 16.1, and 27.2, the discrepancy between m_c and m_f lies within 3 or 4 per cent of the steam supplied to the cylinder. The large-compression engines, Nos. 1 to 12, show far wider differences, especially the compounds, this being in accord with what is set forth in § 22 (*l*) and (*m*); see also the discussion of compression in § 23.

Cases and causes of m_c being much too large relative to m_f are, tests 11 and 19, jacket steam included; test 23, leakage; test 40, excessively wet steam. For the rest, m_c is consistently smaller than m_f

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with jackets, but not in any such regular fashion as to suggest a formulable relation. With high superheat, comparison would be wholly illogical — consider Fig. 139.

Some useful comparisons between m_c and m_f are made in Fig. 138.

PAGE D. THERMODYNAMIC PERFORMANCE.

t_0 Temperature of exhaust, corresponding to the exhaust pressure outside the engine, or to p_0 when marked C in the table. From t_0 can be got, by the use of Table I, the condenser pressure upon which it is based, when this is not directly given. Note that with all the noncondensing engines, t_0 is taken as for the existing pressure of the atmosphere, the engine performance suffering the full effect of back pressure. Compare § 26 (*e*) as to this and the following item.

q_0 Heat in one pound of feed water, at ideal feed temperature, which is higher than t_0 with a jacketed engine. Without jackets or reheaters, q_0 is $(t_0 - 32)$; in a few cases it is so taken with jackets; but generally the engine is credited with the heat returned to the feed by the water from the jackets. The question of ideal and actual feed temperatures is considered in § 26 (*d*) and (*e*). Only in tests 38, 39, and 40 are actual, plant values of q_0 here given.

h_1 Total heat of one pound of steam at pressure p_1 and of the quality fixed by m or s.

h_2 Total heat at the end of adiabatic expansion from p_1 to p_0 — see § 15 (*d*) for this calculation for the Rankine cycle.

Q equal to $(h_1 - q_0)$, is the input of heat per pound of steam, or the heat of formation at p_1 and m or s, above the state of water with q_0.

W_R equal to $(h_1 - h_2)$ is the output of the Rankine cycle, as in § 15 (*d*). For the regenerative cycle, tests 39 and 40, the Carnot cycle efficiency is entered instead of h_2, then under W_R is given the output that would result from this efficiency with the actual input Q — not quite the same as by Eq. (97).

W Actual work output per pound of steam, expressed in B.t.u. It is equal to $(2545 \div S_h)$, as in § 26 (*c*), Eq. (136).

E Absolute thermodynamic efficiency, equal to $(W \div Q)$, as in Eq. (137) — compare the summary in § 26 (*c*).

E_R Relative efficiency $(W \div W_R)$, as in Eq. (138) — see § 26 (*c*) and (*k*).

Q_m Heat supplied per i.h.p. per minute, equal to $(42.4 \div E)$ — see § 26 (*i*).

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TABLE 13, page A. TESTS OF VARIOUS

No.	Condition.	*N*	*V*	*e*	*r*	Diameters	× Stroke.	Ratio.
1	TA1NF	308	719	.31	2.5	8	×14	.....
2	TA1NF	354	531	.28	2.2	9.5	× 9	.....
3	TA1NF	246	533	.12	5.0	14.5	×13	.....
4	TA2NF	293	635	.39	6.3	11.5, 18.5	×13	2.61
5	TC2NF	299	648	.38	6.4	11.5, 18.5	×13	2.61
6	TA1NF	80	400	.30	2.8	21	×30	.....
7	TA2NF	80	427	.55	3.5	23, 35	×32	2.34
8	TA2NF	160	632	.44	5.4	14.2, 23.7	×25.2	2.81
9	TA2NF	240	1037	.50	4.4	15, 25	×26	2.81
10	PA2NF	240	942	.40	5.3	14.2, 26.1	×23.6	2.46
11	TC3JF	72	574	.50	9.9	29.4, 44, 70	×48	5.70
12	TC3JF	61	398	.37	14.7	22, 34, 57	×39	6.75
13	TC4NC	78	700	.....	.....	29, 41.5, 59, 84	×54	8.45
14.1	TA1NC	88	438	.21	3.8	17	×30	.....
2		86	430	.11	5.9			
3		8.6	43	.15	4.9			
4		62	309	.47	1.9			
15.1	TA1NF	406	406	.22	3.8	14	× 6	.....
2	2	401	401	.31	5.0	10, 14	× 6	1.97
3	3	405	405	.61	5.9	7, 10, 14	× 6	4.10
16.1	TA1NF	51	505	.22	4.1	28	×60	.....
2	C	52	520	.12	6.9			
17.1	PA1NF	153	612	.34	2.9	17	×24	.....
2	C	155	620	.26	3.6			
18	TC1NF	13.7	219	.30	3.0	36	×96	.....
19	PC2JF	70	700	.24	12.0	28, 48	×60	3.04
20	PC2HF	77	767	.31	11.5	28, 54	×60	3.69
21	TC2HF	75	752	.33	11.9	28, 56	×60	4.06
22.1	PC2JHM	100	928	.29	14.4	29, 60	×56	4.33
2		100	935	.11	33.1			
23	PC2JHC	102	816	.21	18.9	23, 48	×48	4.38
24	PC2NM	65	783	.30	11.1	30, 56	×72	3.48
25	TC2JHC	34	340	.17	19.9	17, 34	×60	4.18
26	TC2JHF	121	844	.25	15.0	20, 40	×42	4.03
27.1	PC2NC	75	750	.21	18.4	42, 86	×60	4.30
2	TA2NC	75	750	.40	9.2			
3		75	750	.60	7.8			
28	TC2JHF	61	726	.26	22.8	18, 44	×72	6.40
29.1	PC2NF	80	641	.29	20.6	16, 40	×48	6.35
2	JH	80	641	.24	24.5			
30.1	PC2HF	102	716	.32	9.3	16, 28	×42	3.13
2	T	102	715	.27	10.7			
31	PC2HC	101	604	.20	11.6	21, 36	×36	2.98
32	TC3NF	76	608	.30	22.0	19, 29, 46	×48	6.00
33.1	TC3JHF	83	703	.....	.....	34, 49, 61–61	×51	6.40
2	P	83	703	.....	.....			
34	TC2JF	20.2	121	.42	9.1	21, 42	× 36	4.04
35	TC2JHF	18.6	372	.20	20.2	27, 54	×120	4.02
36	TC3JHF	20.3	203	.35	19.6	28, 48, 74	× 60	7.11
37	TC3JHC	21.2	212	.33	22.6	29, 52, 80	× 60	7.66
38.1	TC3JHC	24.1	241	.....	.....	30, 56.5, 84	× 60	7.85
2	P	24.0	240					
39	TC4JHC	36.5	255	.52	16.9	19.5, 29, 49.5, 57.5	×42	8.75
40	TC4JHC	56.9	455	.62	22.4	14.5, 22, 38, 54	×48	14.19

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STEAM ENGINES. TABLE 13, page B.

No.	Clearances.			References, etc.	Fig.
				ENGINES WITH LARGE COMPRESSION. SMALL HIGH-SPEED ENGINES.	
1		.14		No. 23b	
2		.141		16b	
3		.10		13 — From ENGINE TESTS, G. H. Barrus.	
4	.07		.10	42b	
5	.07		.10	41b	
				LOCOMOTIVES.	
6		.093		Nos. 205, 6, 8 — Penn. Railroad Tests, St. Louis Exposition of 1904. See Fig. 103. Mean of two or three tests in each case.	133
7	.167		.057	308, 9	134 I
8	.133		.098	507, 8	134 II
9	.183		.066	609, 10, 11	
10	.117		.114	709, 10, 11	
				MARINE ENGINES.	
11	.124		.080	S. S. *Meteor* — Comm. Inst. M. E., see reference Fig. 135.	135 I
12	.124		.076	S. S. *Iona*	135 II
13				S. S. *Saxonia* ENGINEERING, 1902 I, 326	
				ENGINES WITH SMALL COMPRESSION. SIMPLE ENGINES AND SPECIAL TESTS.	
14		.071		Denton and Jacobus tests, Figs. 86, 90, 98.	86
15	.125	.105	.058	Willans tests, first series, see Fig. 96.	96
16		.03		No. 20b and a — From ENGINE TESTS, G. H.	
17		.02		10b and a — Barrus.	
18		.058		Isherwood, S. S. *Michigan*, see Figs. 91, 92.	91
				COMPOUNDS, CORLISS, AND EQUIVALENT TYPES.	
19	.025		.025	No. 43	
20	.04		.06	48b — From ENGINE TESTS, G. H. Barrus.	
21	.031		.043	55	
22	.028		.04	L. S. Marks — Engs. C to F, 4 tests at full load.	97
23	.02		.03	See Fig. 97 — Eng. B, 2 tests at full load.	
24	.026		.036	J. E. Denton, Trans. A. S. M. E., 15–882.	136 A
25	.035		.027	M. Longridge, ENGINEERING, 1896 I, 132	136 B
26	.047		.070	D. S. Jacobus, see Fig. 88.	88
27	.098		.048	H. G. Stott, Jour. A. S. M. E., Mar., 1910.	137
				HIGH-RATIO COMPOUNDS.	
28	.023		.018	No. 47b, c, d, Barrus' ENGINE TESTS.	
29	.025		.020	G. H. Barrus, ENG. RECORD, 1902 II, 436.	
				HIGH-SUPERHEAT AND TRIPLE-EXPANSION ENGINES.	
30	.041		.058	D. S. Jacobus, Trans. A. S. M. E., 25–264.	139
31	.087		.094	M. Longridge, ENGINEER, 1905 I, 546.	
32	*.03*		*.03*	ENGINEERING, 1894 II, 230	
33	*.03*		*.03*	Zeit. Ver. d. Ing., 1900, 606.	
				PUMPING ENGINES.	
34	.036		.025	R. C. H. Heck,	129
35	.016		.015	F. W. Dean, Trans. A. S. M. E., 16–169,	130
36	.014		.008	R. C. Carpenter, ENG. RECORD, 1893, Dec. 2.	131 I
37	.017		.025	W. F. M. Goss, Trans. A. S. M. E., 21–793.	131 II
38				Bull. Holly Mfg. Co., Engine at Louisville, 1909.	
				ENGINES WITH REGENERATIVE HEATERS.	
39	.013		.004	R. C. Carpenter, ENG. RECORD, 1899 I, 495.	140
40	.060		.035	O. P. Hood, Trans. A. S. M. E., 28-705; air compressor, Michigan copper mine, 1906.	

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TABLE 13, page C. TESTS OF VARIOUS

No.	m, s,	p_1	p_0	p_m	H	S_h	j	S_{l_0}	S_{l_2}	m_c	m_f
1	*0*	97	15.5E	34.9	32	31.20	.....	22.5	21.8	.280	.24
2	.03	107	15.0E	30.8	35	32.99	.....	21.5	22.9	.348	.25
3	*.01*	117	15.5E	20.1	53	32.67	.....	17.7	22.6	.461	.32
4	.004	143	16.0E	30.3	153	25.20	.....	15.6	16.7	.383	.16
5	.004	145	3.6E	37.6	197	19.10	.....	12.0	12.8	.371	.18
6	.005	216	18.6E	91.4	757	25.12	.....	18.7	19.2	.257	.232
7	.016	223	19.4E	79.8	987	19.86	.....	16.8	15.3	.150	.127
8	.012	218	18.7E	49.5	877	20.14	.....	15.5	15.0	.233	.179
9	.013	222	21.4E	49.7	1528	20.43	.....	15.3	14.6	.249	.128
10	91°	203	17.8E	33.0	719	17.61	.....	15.0	14.8	.148	.217
11	*.015*	153	2.7c	29.9	1994	14.98	?	11.6	11.3	.229	.135
12	*.015*	180	0.7c	21.1	645	13.35	.043	8.9	8.3	.335	.217
13	*.015*	207	2.3c	38.8	9099	13.47	.....	.....	.....	.....	.....
14.1	*.01*	105	14.4c	40.9	119	26.47	.....	20.6	21.6	.223	.248
2	*.01*	106	14.4c	26.2	75	27.97	.....	18.9	24.3	.324	.320
3	*.01*	109	14.4c	42.2	12	38.67	.....	16.7	22.4	.568	.591
4	*.01*	43	14.4c	19.2	39	45.29	.....	36.8	38.5	.188	.192
15.1	*.01*	106	14.7c	38.9	34	26.00	.....	18.3	20.4	.296	.232
2	*.01*	159	14.7c	46.0	40	19.19	.....	15.9	15.4	.172	.150
3	*.01*	187	14.7c	45.0	40	18.70	.....	16.6	15.9	.112	.085
16.1	.012	80	16.2E	23.8	452	30.16	.....	21.8	23.2	.278	.246
2	.012	83	3.6E	23.1	444	23.00	.....	14.1	14.4	.385	.358
17.1	41°	90	15.7E	36.8	310	25.64	.....	20.8	20.4	.189	.200
2	16°	95	4.6E	39.4	336	20.51	.....	15.8	15.5	.229	.244
18	*.015*	36	2.2c	19.9	134	35.20	.....	18.3	20.4	.296	.323
19	44°	135	2.5E	26.1	1017	13.26	?	10.1	10.2	.238	.188
20	13°	136	4.0E	23.9	1252	14.01	.040	11.2	11.6	.169	.161
21	*.005*	166	3.2E	30.7	1714	13.27	.035	10.8	10.8	.153	.138
22	86°	177	2.3E	28.4	2220	11.91	.066	10.0	10.4	.08	.161
23	15°	171	1.9E	24.1	1075	13.59	.070	8.4	11.6	.33	.194
24	14°	138	2.1c	25.6	1592	13.50	.....	11.8	11.4	.125	.179
25	0	149	0.5c	23.3	216	13.05	.119	10.0	11.6	.235	.358
26	.007	163	1.0c	26.7	853	12.33	.099	9.4	.....	.16	.19
27.1	9°	190	0.94c	26.0	6846	12.59	.....	10.4	12.6	.176	.136
2	.013	199	16.1c	27.7	7261	18.06	.....	14.6	13.6	.188	.098
3	.010	195	12.1c	37.7	9904	16.13	.....	14.1	15.3	.118	.093
28	.006	166	2.1c	20.5	692	12.67	.077	9.2	9.6	.195	.198
29.1	41°	186	0.9c	23.3	567	11.29	.....	10.0	8.5	.114	.206
2	41°	184	1.2c	23.3	566	11.49	.135	8.7	8.6	.124	.222
30.1	375°	157	1.5c	31.8	420	9.56	0	.....			
2	*.01*	160	2.3c	30.8	407	13.84	?	.....			
31	398°	132	1.0c	22.5	415	9.00	0	(Mean of tests 1 and 3)			
32	.01	178	2.3E	25.8	785	11.75	.....	.....			
33.1	0	218	1.1c	25.2	2976	11.75	?	.....			
2	213°	213	1.1c	24.1	2850	9.62	?	.....			
34	.009	97	2.0c	27.7	281	15.63	.123	11.9	13.1	.130	.292
35	.006	152	1.0c	25.0	643	12.22	.167	7.8	9.7	.233	.338
36	.011	136	1.2c	21.7	574	11.80	.093	9.4	9.0	.128	.228
37	.010	169	1.6c	24.4	783	11.50	.062	8.7	9.3	.195	.226
38.1	.029	174	1.16c	23.9	940	10.82	.075	.....	.....	.....	.....
2	110°	170	1.16c	23.6	926	9.65	.048	.....	.....	.....	.....
39	.013	214	1.2c	35.4	712	12.42	.165	9.7	7.9	.065	.103
40	.057	257	1.3c	31.3	990	11.92	.103	9.3	7.0	.133	.099

Italic figures mean values assumed by author.

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STEAM ENGINES — *Continued.* TABLE 13, page D.

No.	t_0	q_0	h_1	h_2	Q	W_R	W	E	E_R	Q_m
1	212	180	1177	1041	997	135.9	81.0	.081	.596	523
2	212	180	1161	1020	981	140.8	77.2	.079	.548	539
3	212	180	1181	1031	1001	149.9	77.9	.078	.520	545
4	212	180	1189	1024	1009	164.9	101.0	.100	.613	424
5	130	98	1190	915	1092	274.3	133.2	.122	.486	348
6	211	179	1200	1004	1021	196.0	101.3	.098	.517	432
7	211	179	1200	1002	1021	198.2	128.1	.125	.646	339
8	211	179	1200	1003	1021	196.5	126.3	.124	.643	344
9	211	179	1200	1002	1021	198.1	124.6	.122	.628	348
10	211	179	1254	1049	1075	284.8	144.4	.134	.707	316
11	138	121	1181	917	1060	264.3	170.0	.160	.643	265
12	90	70	1184	844	1114	340.1	190.7	.171	.560	248
13	132	100	1187	895	1087	291.5	189.0	.174	.649	244
14.1	211	179	1179	1035	1000	143.5	96.2	.096	.671	442
2	211	179	1179	1035	1000	142.1	91.0	.091	.640	466
3	211	179	1179	1033	1000	146.0	65.8	.066	.451	645
4	211	179	1161	1082	982	79.1	56.3	.057	.711	741
15.1	212	180	1179	1036	999	142.8	97.9	.098	.686	433
2	212	180	1186	1014	1006	171.8	132.6	.132	.722	322
3	212	180	1189	1006	1009	183.0	136.1	.135	.745	314
16.1	212	180	1171	1049	991	122.3	84.4	.085	.690	498
2	146	114	1172	959	1058	213.5	110.6	.105	.518	406
17.1	212	180	1207	1045	1027	162.3	99.3	.097	.611	439
2	143	111	1195	968	1084	226.5	124.1	.114	.549	371
18	130	98	1153	972	1055	180.6	72.4	.069	.401	619
19	128	96	1218	941	1122	277.7	192.0	.171	.691	248
20	114	92	1200	934	1108	266.2	181.6	.164	.682	259
21	116	93	1191	889	1098	302.3	191.9	.175	.635	243
22	130	114	1248	947	1134	300.9	213.8	.189	.711	225
23	109	95	1204	890	1109	314.1	187.5	.169	.597	251
24	128	96	1201	867	1105	277.6	188.5	.170	.679	249
25	80	80	1194	845	1114	348.5	196.3	.176	.563	240
26	102	96	1189	868	1093	321.1	206.4	.184	.625	231
27.1	100	68	1204	868	1137	336.4	202.3	.178	.601	238
2	217	185	1187	1007	1003	180.1	140.9	.140	.768	302
3	202	170	1190	992	1020	198.0	157.7	.155	.797	274
28	120	106	1190	895	1084	295.5	201.0	.186	.683	229
29.1	99	67	1224	895	1157	329.2	225.6	.195	.685	217
2	108	112	1226	893	1114	330.4	221.5	.199	.670	214
30.1	116	84	1391	1008	1307	382.3	266.2	.204	.697	208
2	132	100	1186	910	1086	276.6	183.7	.169	.665	250
31	105	73	1393	1004	1320	389.1	282.8	.214	.727	198
32	102	70	1188	864	1118	324.5	216.6	.194	.667	219
33.1	105	*93*	1200	864	1107	336.2	216.5	.196	.644	217
2	105	*98*	1319	937	1221	382.6	264.6	.217	.693	196
34	126	110	1177	932	1067	245.2	162.8	.153	.664	278
35	102	112	1190	870	1078	316.3	208.3	.194	.659	219
36	108	88	1183	872	1095	299.0	215.7	.197	.721	215
37	118	120	1187	909	1067	299.3	221.3	.207	.739	205
38.1	107	89*	1171	860	1082	311.6	235.1	.217	.755	195
2	107	93*	1260	920	1166	340.1	263.8	.226	.775	187
39	107	281*	1189	(.326)	908	296	204.8	.226	.693	188
40	110	305*	1155	(.334)	850	284	213.5	.252	.753	168

* From actual feed temperatures.

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Note that the ideal output W_R is determined for the actual exhaust pressure (and temperature), not for an assumed mean as suggested in § 26 (*l*): the resulting erratic variation in E_R is strikingly apparent in the comparison of test 12 with 11 and 13, and of 25 with 24. The paired tests 5–6, 16, and 17, with 27, show how much more effectively the engine can utilize theoretically available steam work in the range above atmosphere than in the range of low pressures and large specific volumes — compare § 26 (*k*) and (*m*). Tests 6 to 10 enforce this idea, for even with the large kinetic and other losses shown in Figs. 133 and 134, the locomotive has almost as good a relative efficiency as have the high-grade, multiple-expansion, condensing engines.

(*k*) DIAGRAMS OF PLANT PERFORMANCE. — A very good way of showing the distribution of heat in a steam plant is exemplified in Fig. 141. This graphical heat balance is called the Sankey diagram, having been devised by Captain H. R. Sankey and first described in the Report of the Committee of the Institution of Civil Engineers, "On the Thermal Efficiency of Steam Engines," in 1898.

The scheme is so self-evident and the representation so luminous that no description or explanation seems to be called for. The small difference between 100 per cent of heat received and 22 per cent of work plus 75 per cent of heat rejected to the condenser is the sum of all the radiation losses: these are represented by single lines curving off from the main stream at each section of the apparatus.

In order to correlate the working of the combined boiler and engine plant with the general ideas set forth in § 10, the temperature-entropy diagrams in Fig. 142 have been laid out. Probable and typical proportions are assumed, and to make the example less abstract, numerical values are given in the description which follows. The drawing is purely illustrative, however, the method not lending itself to exact representation of plant action: to get closely determinative data for the heat curve BHE would be almost impossible.

The first diagram, ABEF, shows in simplified fashion the action of the products of combustion in serving as an intermediary between the thermo-chemical operation of heat generation and the purely physical absorption of heat by the water in the boiler. The assumed data are, a calorific power of 14,000 B.t.u., an evaporation of 10 lb. of water, and 20 lb. of hot gases, all per pound of combustible. Then per pound of steam made we have two pounds of chimney gases and 1400 B.t.u. of heat generated.

Starting at 530 deg. abs., point B, and using a constant specific heat of 0.24 for the gases, the impartation of 1400 B.t.u. under constant pressure would raise the temperature by $1400 \div (2 \times 0.24) = 2920$

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deg., or to the ideal height 3450 deg. Actually, because the specific heat increases with the temperature, and because some heat is radiated

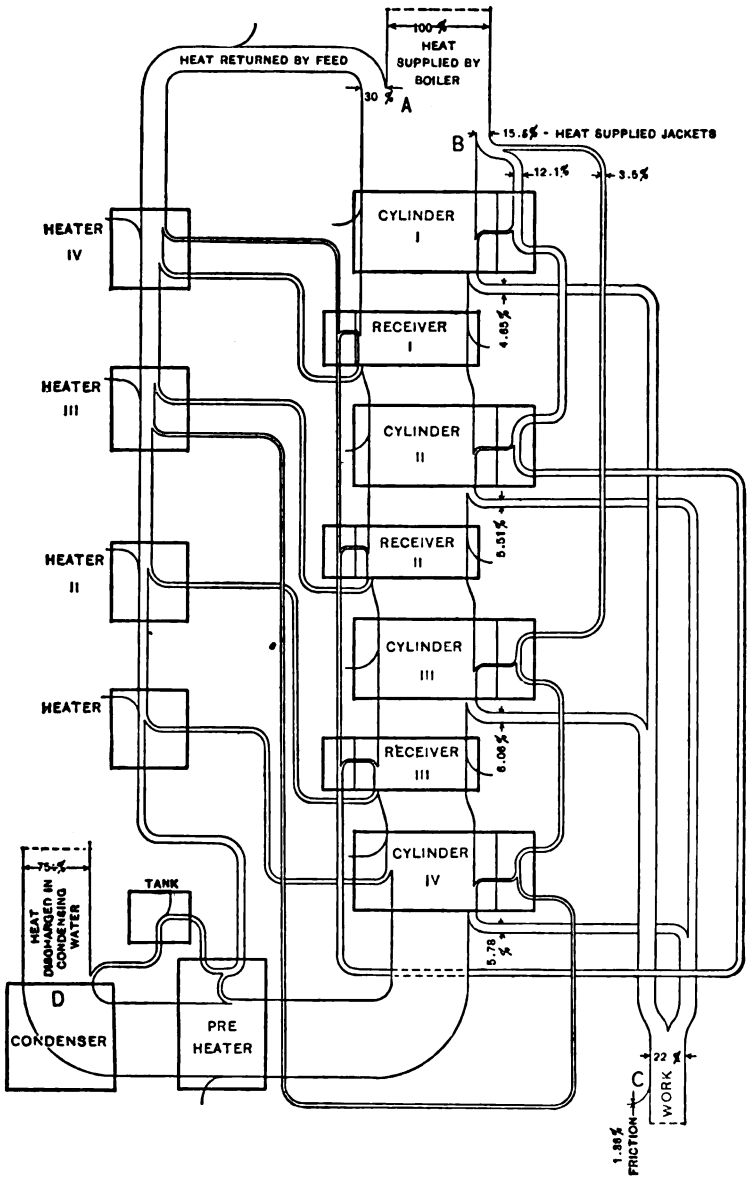

FIG. 141. — Diagram of Heat Distribution, for test A in Fig. 140. From Professor Thurston's paper, Trans. A. S. M. E., Vol. 21, page 216.

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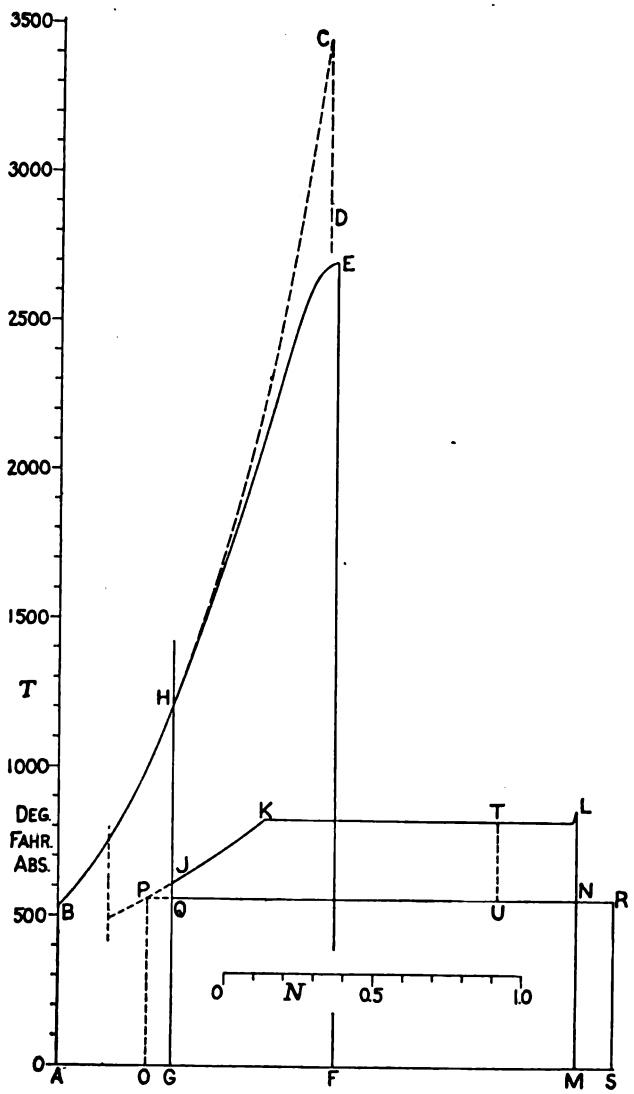

FIG. 142. — Temperature-entropy Diagram for the Whole Plant.

from the incandescent fuel and flame directly to the boiler surfaces, the resultant effect is as if the gases received heat according to some such curve as BHE: 2700 deg. abs. is a good, high furnace temperature, only exceeded with strong forced draft. Of course, this curve by no means shows the real detail of the very complex heat interchanges. Curve BC is got with the help of Eq. (48), and the area ABEF is then made equal to that under BC.

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As the hot gases circulate over the heating surfaces, they give up heat, and may be thought of as cooling along the curve EH. At some terminal temperature GH this heat surrender will cease, and the products of combustion will go to the chimney. The residual temperature of 1200 deg. abs. is rather high, but the residual heat HGAB is only 22.2 per cent of the heat of combustion: for simplicity, all the losses in the boiler are lumped together as heat carried off to the chimney. The absorption of the heat GHEF, 1100 B.t.u. in amount, is the next thing to be considered.

Assuming a strong regenerative action by the jackets of the engine — compare the concluding paragraph of Art. (*i*) — the feed temperature is taken as 150 deg., against an exhaust temperature of 100 deg.; then GJ is 610 deg. abs. Evaporation takes place at 366 deg. or about 165 lb. abs., and a little superheating is produced by the 1100 B.t.u. imparted. The ideal output QJKLNQ is about 326 B.t.u., which would give an absolute efficiency of 0.296; assuming an actual efficiency of 0.215, the relative ratio E_R is 0.726, so that $0.274 \times 326 = 89$ B.t.u. of the theoretically available heat fails to be converted into work. The line TU (not at all a line of operation) divides off this amount; and added to the heat rejected at the exhaust temperature, it augments the latter by the amount MNRS.

The main purpose in drawing this diagram is to show how increase of entropy from heat source to heat receiver is a measure of incurred waste. In the fire and hot gases as source, heat GHEF is carried by the small entropy GF, and only the area QF is below the level of thermodynamic activity. In the highly irreversible impartation of heat to the relatively cool water, there is the great increase of entropy from GF to GM, and the unusable heat grows from QF to QM. Imperfections of the engine cause the further addition NS.

The cycle diagram of the gas engine lies along a curve analogous to BE, but its vertical width is kept comparatively small by the fact that the exhaust curve is of generally similar form to BE: in other words, there is no such uniform low temperature of heat rejection as with the steam engine.

§ 28. Friction and Machine Efficiency

(*a*) Various Conditions as to Output. — In § 3 (*a*) the various kinds of load that an engine may carry have been briefly described. To the class of "power" engines, with rotary load, the term brake horse-power — see § 19 (*n*) and § 26 (*a*) — is most directly and exactly applicable: the work against friction, which absorbs a part of the indi-

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cated work of the steam upon the piston, belongs strictly to the engine itself. This is essentially true even with a direct-connected electric generator, for if the magnetic field of the latter be not excited its rotor is nothing more than a part of the fly-wheel mass of the engine; and the combined unit has practically no more friction of shaft bearings than the engine alone would have. With small and moderate-sized engines, the real output can be directly measured, by means of some form of brake or absorption dynamometer. With electrical loading, the output of the generator is readily known; and if the losses in the generator have been determined, the mechanical efficiency of the engine can be calculated.

Engines of the directly-loaded class generally have the working machine so closely incorporated with them that the friction of the whole combination is the quantity in which we are interested. In the locomotive, tractive friction, due to the weight load on all the axle bearings and augmented by the rolling resistance of the wheels, is not separable from the friction of the engine proper: the useful output is determined by means of a traction dynamometer, which in a laboratory can be placed right behind the locomotive, but on the road must be between the "tender" and the train. With pumps, air compressors, and the like, indicators on the water or air cylinders give the readiest means of measuring the work output — see also § 26 (*j*) as to the calculation of duty.

(*b*) FRICTION LOAD AND POWER. — If indicator diagrams be taken from an engine which is running light, or without external load, their mean effective pressure represents a driving effect just sufficient to overcome the friction of the machine; and from it may be calculated the friction horse-power. A question at once arises as to how this no-load friction compares with the difference between indicated and net power under working conditions. As an observed fact, there is commonly little increase of internal resistance with load. Rationally, the relation between friction m.e.p. or p_f and total m.e.p. or p_m depends upon the kind of friction in the machine. With very slight lubrication, the resistance to motion is almost proportional to the force which presses the rubbing surfaces together, or the coefficient of friction is nearly a constant. With complete lubrication, the surfaces are kept apart by a film of oil, and the resistance to relative motion acts within this fluid: the size of a bearing and the "strength" or required "body" of the lubricant having been determined by the heaviest force to be carried, the "friction" of the joint decreases very little as the external load force is removed.

In an engine, the friction work of the piston and valves and of their rods is likely to increase slowly with the m.e.p.; but the mechanism

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friction will be very much of the second type just described. With well-balanced valves and flooded lubrication, the friction pressure p_f will be almost independent of p_m; with high unit pressure in the bearings and with a scanty supply of oil, p_f will increase with p_m, but always in less than direct proportion.

(*c*) Typical Relations as to friction and efficiency are illustrated in Fig. 143. The base is indicated power, expressed in terms of rated power as unity, the ordinate is effective power to the scale at the left, or efficiency to the scale at the right. Lines or curves like AA and AB show actual output; and when the line OM of equal power is drawn, the vertical intercept gives the lost work or friction horse-power. With AA this difference is constant, with AB it increases at a moderate rate. The efficiency, got by dividing the ordinate of the load line AA or AB by that of the power line OM (or by the abscissa), is shown in the curves AC and AD for the respective cases. This ratio increases very rapidly with small loads, then more slowly; and with heavy loads the curve tends to become horizontal.

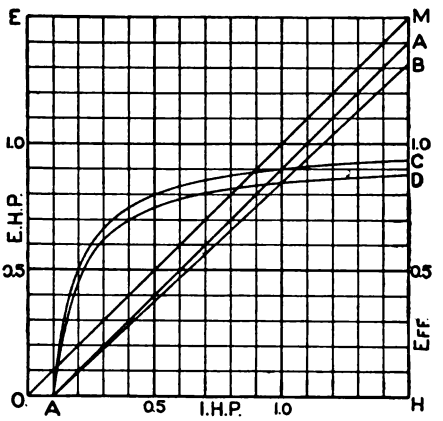

Fig. 143. — Efficiency Diagram.

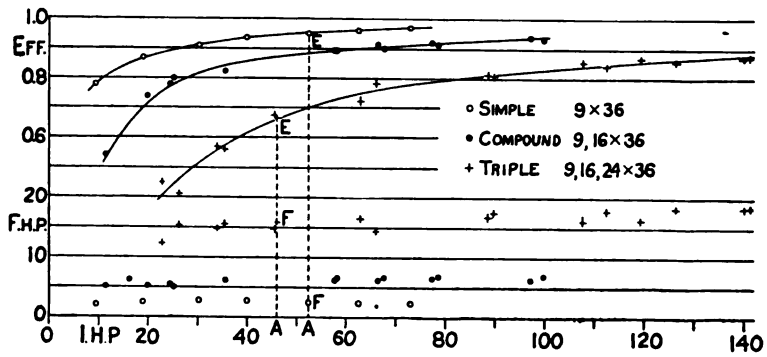

Fig. 144. — Machine Efficiency of Sibley College Engine: see reference under Fig. 95. Whole engine rated 150 h.p. at about 90 r.p.m. Output measured by brakes.

(*d*) Examples of Performance. — In Fig. 144 the scheme of Fig. 143 is modified so as to give a more compact diagram. The first ordinate AF is friction horse-power, which is laid off directly from the base

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line, instead of being an intercept between inclined lines: it is nearly constant for each condition of working, but develops a tendency to increase with the load as the engine is made more complex. The efficiency, AE, is laid out exactly as in Fig. 143.

This laboratory engine is separable into three units, which can be run independently or combined as desired. From simple to compound there is just a little more than a doubling of the friction load, which seems reasonable; but when the three sections are joined together the lost work increases abnormally. The special reason for this does not appear, but the example must be taken as an extreme, not a typical, case of the effect of greater complexity.

The machine efficiency of stationary engines in proper condition ranges from 0.80 to 0.95, with 0.90 as a good value to assume in making approximate estimates.

(*e*) COMBINED EFFICIENCY. — Some examples of the combined working of engines and generators are given in Fig. 145. For com-

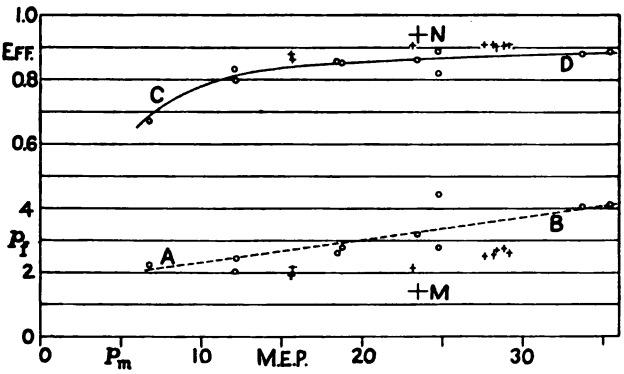

FIG. 145. — Combined Mechanical and Electrical Efficiency, tests reported by L. S. Marks, reference under Fig. 97. Power quantities represented by mean effective pressures. Principal group (circled points) from engine B, other points from engines C, D, E, and F.

paring machines of differing size and speed, m.e.p. is a better quantity to use than horse-power. The base is m.e.p. referred to the low-pressure piston, and the ordinate p_f represents all the losses of power from pistons to generator terminals. At M is given, for engine B, the mean pressure for friction alone, got by indicating the engine when running light. This may be a little low for full-load conditions, and the machine efficiency at N correspondingly high; but the increase of total p_f with load, shown by the rise of the line AB, is principally due to the generator, since the electrical losses have a decided and definite law of increase with output.

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Generator efficiencies commonly range from 0.90 to 0.97 for good machines at rated load, tending to rise with the size of the unit. Using the higher part of the range of mechanical efficiency, say from 0.85 to 0.95, we get 0.80 and 0.92 as the limits between which the combined value is likely to lie — the total efficiency being, of course, the product of the partial efficiencies.

This factor must be taken into account when comparing steam-rate performances per indicated horse-power-hour and per kilowatt-hour, or S_h with S_k as defined in § 26 (*a*). For equal powers the ratio is 0.746, so that

$$S_h \div 0.746 = S_k \quad \text{and} \quad S_k \times 0.746 = S_h. \quad . \quad . \quad . \quad (141)$$

But with a combined efficiency of 0.80 to 0.92, the electrical output from one indicated horse-power will be only 0.60 to 0.68 kilowatt; and this modified ratio will hold between steam per hour per i.h.p. and per kw. A good average value to remember, for use in the transformation represented by Eq. (141), is 0.65 in place of 0.746.

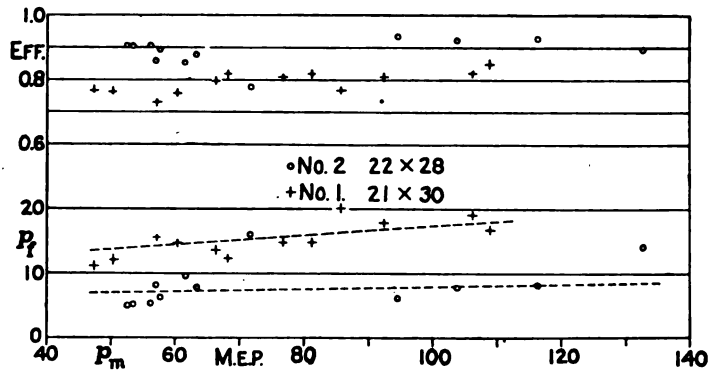

FIG. 146. — Results from Simple Locomotives, Pennsylvania Railroad tests as listed under Fig. 103; steam diagrams from No. 2 are given in Figs. 132 and 133. Base is total m.e.p., ordinate p_f or friction m.e.p.

(*f*) RESULTS FROM LOCOMOTIVES. — The tests represented in Figs. 146 and 147 were made with the locomotives mounted in the usual laboratory fashion, on supporting wheels (one to each driver) which were coupled to friction brakes of the Alden type. The net work is calculated from the pull registered by a traction dynamometer attached directly to the locomotive proper, so that only the friction of the extra axle bearings is included with what strictly belongs to the engine. The scattering of the points is in some degree due to differences in speed, which are not shown in the plots. Mostly, however, it is simply erratic, with the probable error about equally assignable to

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the indicators and to the dynamometer. The exhibit is summed up by naming 0.85 as the average efficiency.

In general, a machine will show less friction at fair speed than at very low speed, chiefly because the bearings run warmer and the lubricant is more fluid; but with very high speeds the friction becomes greater, because of incipient or partial failure of lubrication.

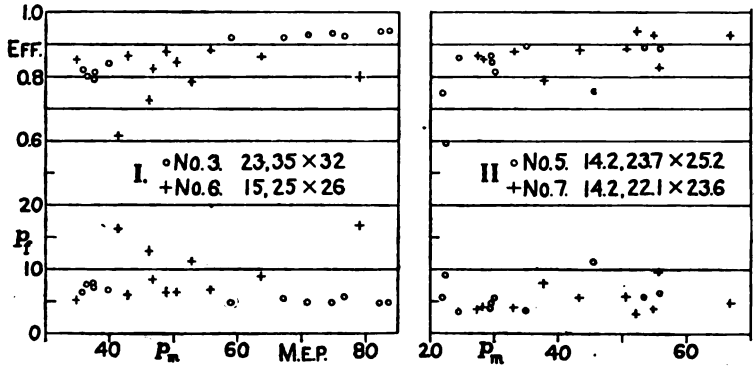

FIG. 147. — Results from Compound Locomotives, same source as Fig. 146. Steam diagrams from Nos. 3 and 5 given in Fig. 134. Base is m.e.p. referred to low-pressure piston.

(*g*) FRICTIONAL M.E.P. — The mean frictional or waste-work pressure p_f has quite a range of variation in the various classes of engines. As a minimum, take from the pumping-engine tests listed in § 27 (*c*) and in Table 13 the average values $E_m = 0.94$, $p_m = 24$, the latter the m.e.p. referred to the low-pressure piston: under these conditions the lost work would be done by $p_f = 0.06 \times 24 = 1.44$ lb. With a rated m.e.p. of 40 lb. and with $E_m = 0.85$, the value of p_f will be 6.8 lb. For stationary engines the common range is perhaps 2 to 5 lb. per sq. in. of the low-pressure piston. For locomotives, with high working m.e.p.'s, the diagrams just given show a common range of 5 to 10 lb. and extreme cases of as much as 20 lb.

This matter has a direct bearing upon the question of the proper degree of expansion, or of the terminal pressure in the engine, as set forth in § 15 (*g*).

(*h*) FRICTION BY FORCE ANALYSIS. — It is obvious that if the variant or average pressure between bearing surfaces has been determined by the methods of the next chapter (§ 34) and if the coefficient of friction is known, the work against friction can be calculated in detail and summed up for the whole machine. This is the logical procedure from the side of machine design, but its quantitative validity is greatly impaired by uncertainty as to the coefficient of friction. For journals

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and bearings the latter is likely to range from 0.02 to 0.06, for the crosshead slide 0.06 to 0.08 is a fair assumption. The internal friction, of piston and valve, is even less determinable. In view of these difficulties, anything more than this suggestion of the analytical method would be out of place here.

§ 29. Proportioning Engine Cylinders

(*a*) Relations Involved. — The problem of determining the size of engine cylinder needed for the development of a certain power is closely related to the calculation of indicated horse-power, as set forth in the first part of § 21. The method is to use Eq. (115), with horsepower H and mean effective pressure p_m as fixed quantities, and choose suitable values of diameter D (or area A), stroke S, and speed N: evidently, the solution is indeterminate, and there is call for an exercise of judgment, guided by a knowledge of practice. For a multiple-expansion engine, the size of the low-pressure cylinder will first be found in this manner, using the reduced or referred m.e.p., of which typical values are given on page C of Table 13: the proportioning of the higher cylinders is another problem.

As a prerequisite to this first step in the design of an engine, the mean effective pressure must be assumed or estimated. Following lines of thermodynamic reasoning, as summed up in § 26 (*c*), we might get the output W per pound of steam by applying an assumed relative efficiency E_R to the ideal output W_R; then choosing a terminal specific volume, such as appears in all the steam diagrams of §§ 22 and 27, we could easily get the m.e.p. by the general method of Eq. (20), page 43. It is better, however, to work from the mechanical side of the subject, calculating the m.e.p. of an "ideal steam diagram" with the expansion curve $pv = C$, like Fig. 57, by the method of § 15 (*i*), then adjusting this by means of an empirical "diagram factor" which takes account of the actual losses of work area.

(*b*) Ideal Diagram and Diagram Factor. — In Fig. 148 the average combined diagram from Fig. 78 — corresponding to Fig. 82, but first laid out, in full line, on the ordinary plan of bringing the clearance lines to the axis of zero volume — is used to illustrate this matter. The top line AB shows steam-pipe pressure p_1, the bottom line RD condenser pressure p_0; and the hyperbola BC is drawn through high-pressure cut-off, with O as origin. One scheme, hich has been quite a good deal used, is to compare the sum of the actual work areas with the full area ABCDR: in this particular example, the ratio is 0.55.

Thus to include the clearance and compression areas within the

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ideal area hardly seems desirable. The opposite extreme would be to use the dotted diagrams, with their compression curves brought to the axis OA, as in Fig. 82 and Fig. 75: then the "ideal" expansion curve would be HN, and AHNPR the basal area. This comparison gives a

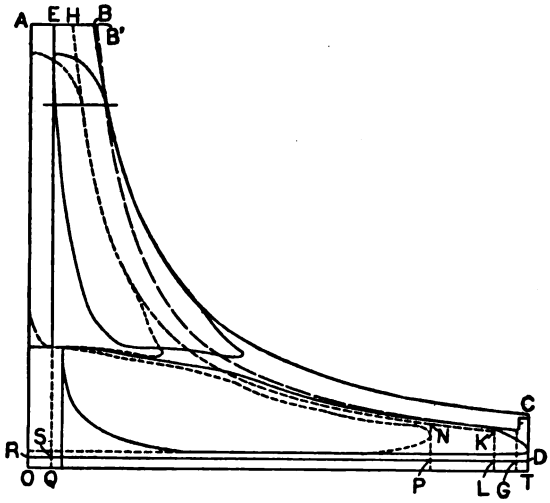

FIG. 148. — To Illustrate the Diagram Factor.

high factor, here 0.84: but beside the extra graphical work involved, it has the great disadvantage that the base-length RP no longer represents the cylinder volume, and hence is not a proper divisor by which to get m.e.p. from area.

A glance at the examples in Table 14 will show that the degree of conformity to such a standard as the ideal steam diagram is low and exceedingly variable. Since the purpose of recorded diagram factors from working engines is merely to serve as a help in making approximate estimates of probable action, simplicity and convenience are of more importance than logical correctness. Disregarding then the clearances, a simple expedient is to back the end lines of the cylinder diagrams up against the vertical axis. In Fig. 148, imagine the low-pressure diagram to be shifted to the left until it touches the line QE, or lay off its length in SG, and draw the hyperbola B'F with Q as origin. The factor is now the ratio of the card areas to EB'FGS, and as worked out in Example 32, it is 0.76 for this figure.

In the *Code of Rules for Engine Tests* of the American Society of Mechanical Engineers, Vol. 24, page 751, the method recommended is the one just described, except that the initial volume, measured off

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from E to B′, is the intercept between the compression and expansion curves when produced up to the steam-pressure line AB, corresponding with GH in Fig. 107. The gain from this slight added complexity is questionable.

EXAMPLE 32. — In the engine and test of Figs. 78 to 82, and of Example 26, page 164, the data for the calculation of ideal m.e.p. and of the diagram factor are as follows:

Steam pressure $p_1 = 111$, condenser pressure $p_0 = 3.9$, cut-off pressure $p = 90$, all in pounds absolute; mean high-pressure cut-off at 0.295 of the stroke, and cylinder ratio equal to 2.36; m.e.p. referred to large piston, $p_m = 25.0$ lb.

Taking the nominal high-pressure volume as unity, the expansion ratio from initial cut-off to terminal volume, or from $v_1 = 0.295$ to $v_2 = 2.36$ is $2.36 \div 0.295 = 8.0$; while the total ratio, from $p_1 = 111$ instead of from $p = 90$ is

$$r = 8.0 \times \frac{111}{90} = 9.87.$$

Substituting this in Eq. (102), we get the ideal m.e.p.

$$p_{mi} = p_1 \frac{1 + \log_e r}{r} - p_0 = 111 \times \frac{3.289}{9.87} - 3.9$$
$$= 37.0 - 3.9 = 33.1.$$

Then the diagram factor f is $25.0 \div 33.1 = 0.756$.

(c) EXAMPLES OF THE DIAGRAM FACTOR. — Data and results from nearly all the steam diagrams in this Chapter, with a few from simple-engine diagrams in Chapter V, are collected in Table 14. The factor f, in the last column, varies widely. The tests may be briefly discussed as follows:

Numbers 1 to 6 are large-compression engines, and have big kinetic losses — compare § 19 (j). In the locomotives, Nos. 2, 3, and 4, the latter include the effect of the high back-pressure caused by the exhaust nozzle. The very low values, 3b and 4b, are due to excessive compression. Typical ranges of f in this class are, for simple engines, 0.80 to 0.85, for multiple-expansion engines, 0.70 to 0.75.

The remaining tests are from engines of the small-compression class, No. 7 simple, 8 to 10 Corliss or equivalent compound, 11 to 14 pumping engines. The normal range of f seems to be from 0.85 to 0.95. In Nos. 9 and 12, the high factor results from strong jacket action, which holds up the value of pv (relative to that at high-pressure cut-off) as the steam expands. In engine 10, the low factor is due to large kinetic losses, together with a very considerable compression effect — see Fig. 137: but in spite of this, the engine has a very high relative efficiency.

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That f exceeds unity in test 7b is due to the assumptions upon which p_{mi} is based. Omission of the clearance volume makes the curve like B'F in Fig. 148 drop so rapidly that on a diagram such as No. 15 of Fig. 90 it will fall below the actual expansion curve. Commonly, this diminution of the ideal area and mean pressure is decidedly overbalanced by the decrease of actual area through compression; but here the compression is very small and the factor comes out too big.

The amount of empirical knowledge really needed for an intelligent choice of the diagram factor to be used in any case is enough to guide the more definite operation of blocking out the steam diagram and sketching in curves which are likely to be quite nearly realized in the engine.

From a glance down the p_m column of Table 13, it appears that the m.e.p. for rated load, referred to the low-pressure piston, ranges from about 25 in stationary engines of the more economical types to 40 in those engines which are required to give a larger output of power per unit of space and weight.

(*d*) DETERMINING SIZE AND SPEED. — Having i.h.p. H and m.e.p p_m, and putting the factor 2 into Eq. (115) for a double-acting engine, we get the equation of relation

$$ASN = 198{,}000\,\frac{H}{p_m},$$

or

$$D^2SN = 252{,}100\,\frac{H}{p_m}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (142)$$

This can be made determinate by fixing N and choosing a ratio between D and S, or by assuming a value of the piston speed $2\,SN/12$. The method of handling the problem of size and speed will best be shown by a couple of examples.

EXAMPLE 33. — Find suitable cylinder dimensions for a high-speed engine, like that in Fig. 2, which is to develop 110 i.h.p. on 36 lb. of m.e.p.

In this type of engine, the diameter and stroke are nearly equal. First lettering $S = D$ and substituting in Eq. (142) we have,

$$D^3 = \frac{252{,}100 \times 110}{N \times 36} = \frac{770{,}300}{N}.$$

Make several solutions with different speeds:

N =	200	225	250
D^3 =	3852	3424	3081
$D = S$ =	15.68	15.07	14.55

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TABLE 14. EXAMPLES OF THE DIAGRAM FACTOR.

No.	Fig.	Condition.	p_1	p_0	p	e	R	r	p_{mi}	p_m	f
1	84	1-14-15-225	120	14.7	90	.333	.....	4.0	57.0	48.0	0.84
2a	133-5	1-21-30-80	218	14.4	190	.171	.....	7.1	77	64	0.83
b	8	80	219	14.4	179	.410	.....	3.0	139	117	0.84
3a	134 I A	2-35-32-80	221	14.4	160	.610	2.34	5.3	102.3	74.8	0.73
b	B	160	223	14.4	160	.431	2.34	7.6	74.5	35.8	0.48
4a	134 II A	2-24-25-160	223	14.4	160	.568	2.81	6.9	80.0	53.3	0.67
b	B	240	234	14.4	160	.322	2.81	11.6	52.0	30.5	0.58
5	135 I	3-70-48-72	153	2.7	110	.663	5.70	12.0	41.7	30.5	0.73
6	135 II	3-57-39-61	170	0.7	110	.455	6.75	22.9	29.8	21.4	0.72
7a	90-1	1-17-30-70	104	14.4	80	.687	.....	1.9	76.0	64.0	0.84
b	15	86	106	14.4	80	.140	.....	9.5	22.0	26.0	1.18
8	136 A	2-56-72-65	138	2.1	110	.311	3.48	14.1	33.7	27.6	0.82
9	136 B	2-34-60-34	149	0.5	130	.199	4.18	24.1	24.4	23.4	0.96
10a	137 A	2-86-60-75	190	1.0	160	.23	4.30	22.2	34.1	26.0	0.76
b	B	75	199	16.1	160	.40	4.30	13.4	37.4	26.4	0.71
c	C	75	195	12.1	160	.53	4.30	10.1	51.9	38.6	0.75
11	129	2-42-36-20	97	2.0	84	.468	4.04	10.0	30.0	26.8	0.89
12	130	2-54-120-19	157	1.0	130	.202	4.02	23.2	26.3	25.0	0.95
13	131 I	3-74-60-20	136	1.2	130	.363	7.11	20.5	25.5	22.1	0.87
14	131 II	3-80-60-21	169	1.6	150	.366	7.66	23.6	28.2	24.0	0.85

For fuller information about the engines, see the figures referred to.

Under CONDITION, the first number shows the stages in expansion (1 = simple, 2 = compound, 3 = triple expansion), the second the diameter of the low-pressure piston, the third the stroke, the fourth the speed in r.p.m.

The other symbols are all defined in Example 32, which illustrates the method of calculating expansion ratio r, ideal mean pressure p_{mi}, and diagram factor f. Pressures p_1, p_0, and p_m, also cylinder ratio R, have the same meaning as in Table 13, where most of these tests appear; but cut-off e is different, corresponding with RB/MN in Fig. 73, while p is the same as OS in that figure.

The determinations for the above table are all made from the diagrams as redrawn for this book; further, in most cases, the conditions belonging to a particular set of indicator cards are not the same as the average conditions for a long test: these reasons account for the small departures from Table 13 which will be noted on close comparison.

Using simple dimensions in inches,

Let	D =	15.5	15	15	14.5
and	S =	16	15	14	15
then	SD^2 =	3842	3375	3150	3154
and	N =	200.3	228.2	244.4	244.1

Which of these solutions to use is a matter of choice.

EXAMPLE 34. — Find the diameter of the low-pressure cylinder of a compound engine which will develop 2000 i.h.p. on 25 lb. of referred m.e.p., with a stroke of 60 in. and at a speed of 75 r.p.m.

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By Eq. (142),

$$D^2 = \frac{252{,}100 \times 2000}{60 \times 75 \times 25} = 4482,$$

and $$D = 66.95 \text{ or } 67 \text{ in.}$$

If the piston speed had been given as 750 ft. per min., the values of S and N would have been open to choice.

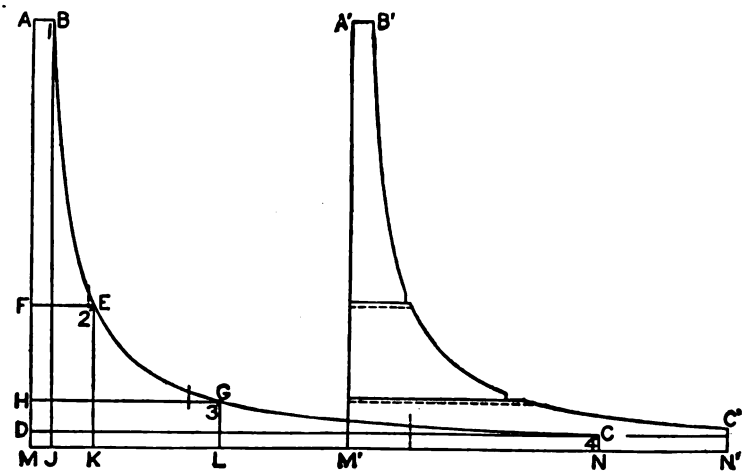

FIG. 149. — A Three-stage Diagram with Complete Expansion.

FIG. 150. — The Introduction of Terminal Drop.

(*e*) THE PROBLEM OF CYLINDER RATIO. — In dividing the total operation among the two or more stages of a multiple-expansion engine, a fairly equal distribution of work among the pistons and the avoidance of excessive loss by incomplete expansion in the higher cylinders, or through receiver drop, are the important objects to be kept in mind. The showing of Fig. 76, where with equal work division there is very nearly the same temperature range in both stages, is so closely typical that, as a general thing, the question of temperature range may be left to take care of itself.

Retaining the ideal conditions of Fig. 76, including a constant receiver pressure, and adding the requirement of complete expansion in all stages, consider the problem of dividing area ABCD, Fig. 149, into three equal parts. With the expansion curve $pv = C$, and using component areas as in § 15 (*i*), we have

$$\text{area ABEF or } A_1 = p_1 v_1 \left(1 + \log_e \frac{v_2}{v_1}\right) - p_2 v_2$$

$$= pv \log_e r_1, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (143)$$

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letting pv stand as a general value for the curve BC. Similarly,

$$\text{FEGH} = A_2 = pv \log_e r_2; \qquad \text{HGCD} = A_3 = pv \log_e r_3.$$

For equal partial areas A_1, A_2, and A_3, the individual expansion ratios must be equal, or

$$\frac{v_2}{v_1} = \frac{v_3}{v_2} = \frac{v_4}{v_3} = r;$$

further, the total ratio R or v_4/v_1 will evidently be the product of r_1, r_2, and r_3: wherefore, in this case, $R = r^3$; or in general, for n stages,

$$R = r^n. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (144)$$

Note that here, as compared with Art. (*b*), it has been convenient to assign new meanings to r and R.

In Fig. 150 the expansions are cut short at the volumes indicated by dotted lines in Fig. 149; and to equalize areas the cut-offs are moved up a little in stages 2 and 3, raising the receiver pressures above the original levels which are dotted in on Fig. 150. Usually, the amount cut away from the last stage is relatively much greater than from the earlier stages.

(*f*) Equalization of Work Areas. — Coming back to the compound engine, as in Fig. 76 and in Fig. 151, and using subscripts cor-

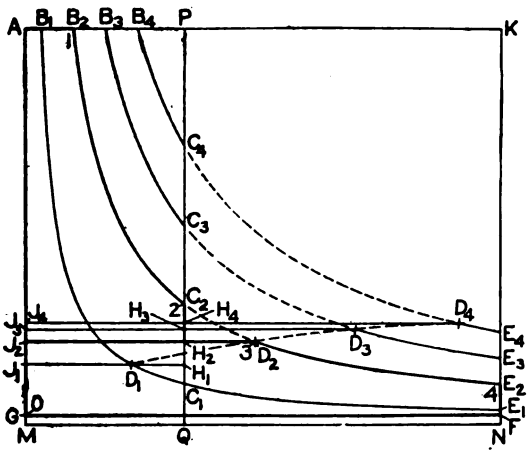

Fig. 151. — Diagrams Showing Equal Division of Work, with variation in the cut-off in both cylinders.

responding with the numbers 1, 2, 3, 4, and 0 on both diagrams, we have the fundamental relation,

$$p_1v_1 = p_2v_2 = p_3v_3 = p_4v_4. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (145)$$

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The areas are,

$$\text{ABCHJ} = A_1 = pv\left(1 + \log_e \frac{v_2}{v_1}\right) - p_3 v_2; \quad . \quad . \quad . \quad (146)$$

$$\text{JDEFG} = A_2 = pv\left(1 + \log_e \frac{v_4}{v_3}\right) - p_0 v_4. \quad . \quad . \quad . \quad (147)$$

Equating and dividing by pv, using the particular values in Eq. (145) as convenient, we get

$$\log_e \frac{v_2}{v_1} - \frac{v_2}{v_3} = \log_e \frac{v_4}{v_3} - \frac{p_0}{p_4}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (148)$$

If there is to be no receiver drop, or if $v_3 = v_2$, and if further we let $v_1 = Ev_4$, this becomes

$$\log_e \frac{v_2}{Ev_4} - \log_e \frac{v_4}{v_2} = 1 - \frac{p_0}{p_4},$$

or

$$\log_e \frac{1}{E}\left(\frac{v_2}{v_4}\right)^2 = 1 - \frac{p_0}{p_4};$$

whence

$$\frac{v_2}{v_4} = \sqrt{E}\sqrt{\text{anti-log}\left(1 - \frac{p_0}{p_4}\right)} = C\sqrt{E}. \quad . \quad . \quad . \quad (149)$$

This E, which may well be called the ratio of total cut-off, is the reciprocal of R above; the cylinder ratio v_4/v_2 is no longer the square root of R, but it is proportional to that root according to a factor which depends upon the ratio of final pressure-drop, p_0/p_4.

Equation (149) is the only application of Eq. (145) that is of real, practical use. It is to be considered as fixing, not the high-pressure cylinder volume v_2, but rather the low-pressure cut-off volume v_3, and with it the receiver pressure. Having thus gotten an approximate value for p_3, the best procedure is to draw the line JD tentatively, choose a v_2 which gives a reasonable receiver drop, then lower JD, by trial and planimeter measurement, until equality is reëstablished.

(*g*) VARIATION OF LOAD. — In Figs. 151 and 152, the heavy-line diagrams marked by the subscript 2 are supposed to represent the best working of the engine, at its proper, rated load. In Fig. 151, the division of work is kept equal as the power of the engine — determined chiefly by high-pressure cut-off — varies over a wide range. To secure equality, the low-pressure cut-off must also change: here D has a wider range on the base MN than B has on MQ, and D is always later in the stroke than B. In this diagram we see that at big loads there is an excessive receiver drop, while in case 1 the expansion curve B_1C_1 sinks below the exhaust line J_1H_1, forming a negative-work loop just as in a simple engine under similar conditions, and with a corresponding loss of efficiency, according to the reasoning of § 15 (*g*). In the whole action, the receiver pressure varies over a comparatively narrow range.

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Fig. 152 shows the other type of relative valve action, the low-pressure cut-off remaining fixed while the high-pressure cut-off changes. The result is a wide variation in receiver pressure, and a decided departure from equality in work division; but the receiver-drop loss is kept down to a very moderate amount with any load.

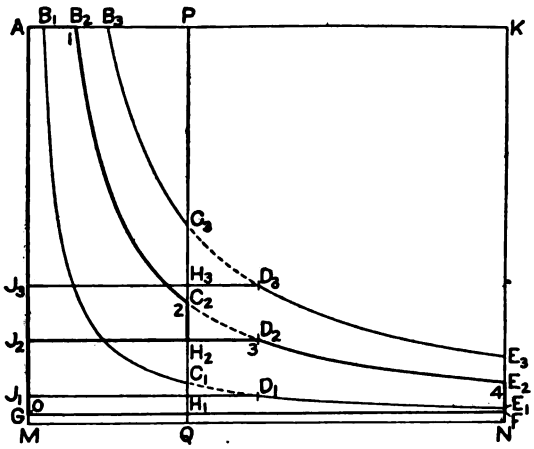

Fig. 152. — Constant Cut-off in the Low-pressure Cylinder.

(*h*) General Relations. — The showing of Arts. (*e*) to (*g*) will now be briefly summed up, for the two-stage engine, with the addition of a few almost self-evident conclusions as to the result of certain other variations in condition. For the cycle of any engine, the fundamental determinants are initial pressure p_1 and exhaust pressure p_0; and the primary variable with load is the first or high-pressure cut-off, upon which directly depend the total expansion ratio $R = v_4/v_1$ and the terminal pressure p_4.

With these leading dimensions of the cycle fixed for a compound engine, the division of work between the cylinders depends chiefly upon the receiver pressure, and this in turn upon the low-pressure cut-off. As the latter is earlier, it raises the admission pressure in its own cylinder, increasing mean effective pressure and share of work; the effect is just the opposite of that produced by a change in the initial cut-off which determines amount of steam admitted.

With the load constant, the work done in the high-pressure cylinder is increased,

(*a*) By making the low-pressure cut-off later.

(*b*) By raising the boiler pressure: this slowly diminishes the product pv, or the amount of steam admitted to perform a given amount

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of total work per revolution; and with a fixed low-pressure cut-off volume v_3, the receiver pressure p_2 is lowered. The addition to the high-pressure diagram because of a higher admission line and a lower exhaust line is greater than the loss from moving the expansion curve inward.

(*c*) By raising the exhaust pressure of the engine: this subtracts more from the bottom of the low-pressure diagram than is added on account of the increase in pv and the consequent rise in receiver pressure.

With variable load and a fixed low-pressure cut-off, the high cylinder takes a larger share of a small load, a smaller share of a large load, as appears from Fig. 152. For equality, the two cut-offs must vary together, but not in exactly the same manner.

(*i*) CYLINDER PROPORTIONS. — The influences taken into account in Art. (*e*) do not fix the cylinder ratio with any exactness, so that there is always a considerable element of judgment in the design of multiple-expansion engines. In Fig. 76, for instance, it is evident that v_2 can be varied over quite a little range without much change in total work or in work division, only the receiver drop showing a large relative variation. The general idea of Eq. (144) underlies the whole matter, but that simple progression is only very roughly followed.

In summary and extension of the information in the R column of Table 13, it may be stated that in practice the overall volume ratio ranges from 2 to 5 in compound and from 5 to 10 in triple and quadruple engines, with 2.5 to 4 and 6.5 to 9 as the regions within which most engines are found. The smallest ratios are used in compounds which need a big high-pressure cylinder for the sake of liveliness in reversing, as in mine hoists and reversing rolling mills. Large-compression engines have smaller ratios between the cylinders than have those with small compression and fuller expansion in the individual stage. The high-ratio compound, represented by tests 28 and 29 in Table 13, is a special type in which the aim is to get very long expansion with but two cylinders: as against the triple arrangement, it saves the kinetic losses of one transfer of steam, but to balance this it has relatively more receiver drop, and perhaps a little bigger cylinder loss.

Limitation of space forbids further exposition of this subject here: a large number of examples, in tabulated form, will be found in STEAM ENGINE, Vol. II, pages 506 to 509.

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CHAPTER VII

WORKING AND CONSTRUCTION OF THE ENGINE

§ 30. Forces in the Machine

(*a*) In This Chapter will be considered the force actions within the reciprocating engine and the motions of its parts, and typical examples will be given of the construction of the members of the main engine mechanism. The discussion of force action begins with steam pressure on the piston and runs through to the final working force, which meets and overcomes the load resistance. In the study of motion, the determining fact is that, through the influence of the fly wheel, the crank shaft is constrained to practically uniform rotation; therefore we reason from the shaft back to the piston.

The total or mean effect of the steam pressure, in doing work upon the piston, has been fully considered in § 21 (*b*) to (*d*): further development along that line has been carried out in §§ 28 and 29. The instantaneous, variable force action in the engine is the principal subject now before us, and the first step is to take a general view of the whole system of working forces. In this, and in the more detailed discussion of later sections, friction is not taken into account, or ideal mechanical working is assumed. Further, the force of gravity on the reciprocating parts is practically omitted: in a horizontal engine its effect in the direction of the main force action is insignificant; in a vertical engine it can be very simply included, in a manner which is noted in § 32 (*c*).

In Fig. 153 all the forces that have to do with work performance are represented graphically, upon an outline of the engine, showing instantaneous conditions for a particular position of the mechanism.

(*b*) Forces on the Piston Slide. — Upon the sliding piece made up of piston, piston rod, and crosshead act forces shown first on the main figure, and then, in combination, at A. These are:

P_F = forward steam pressure, the full, absolute pressure acting on the forward-moving or driving side of the piston at the instant.

P_B = back pressure, on the exhaust or return-stroke side of the piston.

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Then

$P_E = P_F - P_B$ is the effective steam pressure on the piston.

F_1 = inertia force of slide, its reaction against the acceleration essential to its rapidly changing velocity.

$P_D = P_E - F_1$ is then the final effective driving force, delivered at the wrist pin W for transmission to the crank.

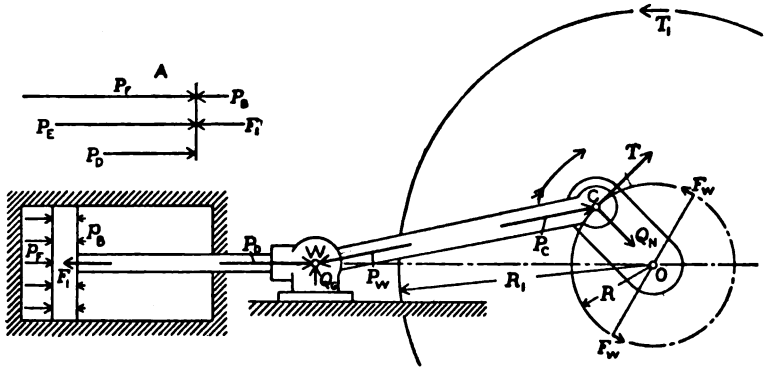

FIG. 153. — Forces on the Moving Parts.

These forces may be expressed either in pounds per square inch or as total forces on the piston, bringing in the area factor A, according to the relation $P = p \times A$. The steam pressures are usually got from indicator diagrams, and it is then more convenient to keep them in pounds per square inch and bring the other forces to the same terms. Here, however, we are considering total effects; so that what are really distributed forces, spread out over the surface of the piston or the mass of the slide, are reduced to and represented by single, concentrated forces along the axis.

From the beginning to, approximately, the middle of the stroke, the slide is being accelerated, and the inertia F_1 points in the direction shown, against the forward driving force; but in the second half of the stroke the slide is retarded (or its acceleration is negative), and F_1 then reverses. At high speeds, this force due to the mass and motion of the working parts modifies very materially the play of the resultant steam pressure P_E.

Considering just the one position shown, the work being done upon the piston by P_E is not all transmitted to the crank, but part of it is being stored in the slide as kinetic energy. Since, however, the slide has zero velocity at each end of the stroke, the inflow and outflow of energy must balance; and, as a net result, all the effective steam work will be carried over to the crank.

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(c) TRANSMISSION TO THE CRANK. — The principal forces in equilibrium at the wrist pin are P_D and P_W, the latter the pressure of the connecting rod upon the pin; but since these are, in general, not in line, the third force essential to equilibrium is supplied by the guide reaction Q_G, perpendicular to the slide bar. In this figure, Q is used to indicate a force which does not move in the direction of its action — that is, one which acts upon a body that does not thus move — so that it cannot do work. A simple trial of conditions during the return stroke (see Fig. 162) will show that with both the rod slant and the force P_D reversed, Q_G will act in the same direction as during the forward stroke.

The connecting rod WC has, of course, an inertia force of its own, which would have to be determined and taken into account in order to find the exact manner of the force transmission. This is a difficult operation, and it is usual to adopt the approximation of considering a part of the mass of the rod concentrated at W, where it adds itself to the slide and increases F_1, and the rest at C, where it has a radial, centrifugal inertia force, with no turning effect upon the crank. Then the rod can be taken as a weightless link, transmitting force along the center-line from W to C.

Coming now to the crank pin, we have the force P_C, equal to P_W reversed, exerted upon it by the connecting rod. Resolving this into components perpendicular to and along the crank arm OC, T is the turning force, or force tangential to the crank circle, while Q_N is the radial component, which simply presses the shaft against the bearings.

This turning force T, with the lever arm R, must equal in effect the resistance T_1, here taken tangential to the wheel, at the radius R_1. In the aggregate the average moment TR must equal T_1R_1; but since T varies widely, equilibrium exists only at certain positions of the engine: in all others there is an unbalanced turning moment, free to give angular acceleration to the shaft, and acting against the moment of inertia of the fly wheel, represented by the couple F_W-F_W.

Under the assumption that no work is lost through friction, the work done by T upon the crank in one revolution must be the same as that done upon the piston by the steam pressure. During each stroke the piston moves through the distance $2R$, the crank pin through πR; using the subscript m to mark mean values, we have

$$P_{Em} \times 2R = P_{Dm} \times 2R = T_m \times \pi R;$$

whence, using P_m for either m.e.p.,

$$T_m = \frac{2}{\pi} P_m = 0.6366\, P_m. \quad . \quad . \quad . \quad . \quad . \quad (150)$$

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(*d*) EXACT FORCE ACTION ON THE CONNECTING ROD. — This is illustrated in Fig. 154, the explanation of which is as follows:

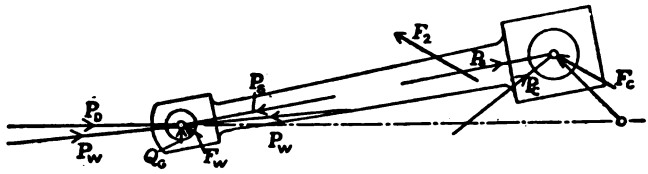

FIG. 154. — Forces on the Connecting Rod.

Assuming the total inertia force F_2 — the concentrated resultant of the small inertias of the distributed particles of the rod — to be known, we may replace it by parallel components at the pins F_W at W, F_C at C. Now besides transmitting force along its center-line, the rod must receive from the pins component forces equal and opposite to F_W and F_C, respectively. Or, conversely, the pressure P_W on the wrist pin will be the resultant of P_S, along the rod, and F_W. By a method the development of which belongs to the graphics of machine forces, it is possible to make the resultant of P_D and Q_G equal that of F_W and P_S: and then carrying this P_S over to C and combining it with F_C, we get the other pin pressure.

For the sake of knowing how nearly correct is the approximation to rod inertia described in Art. (*c*), the inertia effect of the rod was fully worked out, for a typical case, in THE STEAM ENGINE, Vol. I, §§ 37 and 38. It was there established that the scheme of "concentrating" the rod mass in two parts at the pins is amply accurate for all practical purposes. The exact discussion is not reproduced here, for it really belongs to the subject of mechanics of machinery rather than to a study of the steam engine.

(*e*) EQUILIBRIUM OF THE SHAFT. — In Fig. 153 the system of forces acting on the shaft is not complete, because the pressure with which the bearings balance the resultant of all the other forces is not included. The complete set of forces is shown in Fig. 155, several being added to those given on Fig. 153. An important mechanical principle which applies in this case is shown at II, and may be stated as follows:

If the force P acting upon a body does not pass through the center or axis O about which the body is compelled to turn, this force will have two effects: it will tend to push the body straight ahead in its own direction, as though a force B_1 equal and parallel to itself were applied at the center, and resisted by the equal and opposite reaction of the bearing; and will also exert a turning moment, that of the couple made up of P and B_2, tending to turn the body about O.

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In order, then, to find the pressure on the bearing, we have only to combine all the forces acting upon the shaft, whether they pass through the center or not. On Fig. 155 these forces are: —

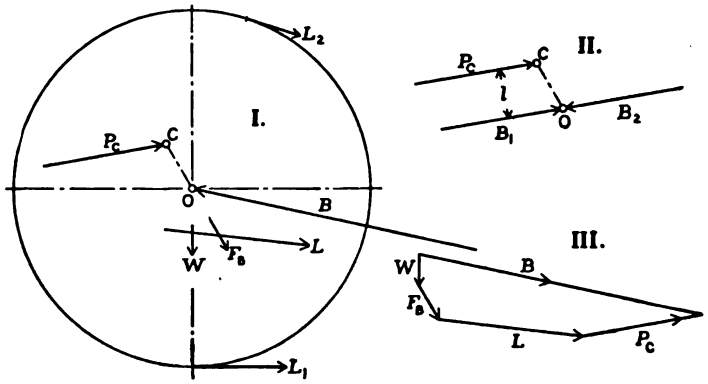

FIG. 155. — Forces on Shaft and Wheel.

P_C = pressure of connecting rod on crank pin.

W = weight of whole rotating piece.

L = resultant load force: here the power is supposed to be taken from the engine by a belt, the difference between the two tensions, L_1 on the tight or driving side and L_2 on the slack side, being equal to the T_1 on Fig. 153. Then L is the resultant of the two belt pulls.

F_B = centrifugal force of the counterweight: this is an eccentric mass, attached to the crank opposite the crank pin, in order that its centrifugal force may partly balance the inertia force F_1 of the reciprocating parts, in a manner which will be explained presently.

The resultant of these four forces is found in the force polygon at III, by laying them out in order and drawing the closing side B: as marked here, its arrow against the others, it is the resultant; as drawn in I it is the equilibrant, the reaction of the bearing against the shaft.

Of these forces, the one that shows the greatest variation in character is the load L. In many cases — as when the engine is direct-connected to an electric generator, or to a screw propeller — the resistance is a simple torque, with no tendency to press the shaft upon the bearing in any particular direction. In a locomotive, the resistance is tangential to the driving wheels. In direct-acting air compressors and pumps, where the whole engine mechanism is used merely to regulate the speed and stroke, no power being taken from the shaft, the only resistance to the turning moment on the crank is the angular inertia of the wheel, again a simple torque.

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With the symmetrical arrangement of the center-crank engine in Figs. 2 and 7, it is strictly correct to consider the forces on the shaft as all acting in one vertical plane. But in some cases this is only a representation of resultant effect: with a side-crank engine, for instance, it is necessary to go through quite a complex analysis in order to determine the forces on the bearings and the stresses in the shaft.

(*f*) THE FORCES ON THE ENGINE BED, on account of steam pressure and of the inertia of the reciprocating parts, are shown in Fig. 156. The resultant of the two steam pressures on the cylinder heads is P_E.

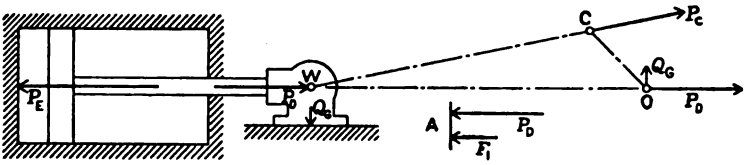

FIG. 156. — Forces on the Engine Bed.

The driving force P_D combines with the guide reaction to give the crank-pin pressure P_C; and this, transferred to the center of the shaft according to Fig. 155 II, and there resolved back into its components, gives the forces Q_G and P_D as exerted upon the bearing (the bed) by the shaft.

Considering horizontal forces, it appears that the bearing pressure P_D is less than the steam reaction P_E, so that the engine bed is not in static equilibrium. Separating P_E into two parts, as at A, one equal to P_D, the other to F_1, we see that the working forces proper do form a balanced system within the machine, as shown by the opposing P_D's; but the absorption of a part of P_E-on-the-piston in accelerating the latter, or overcoming the inertia F_1, leaves the corresponding part of P_E-on-the-cylinder as an unbalanced force, free to accelerate the whole body of the engine in the other direction. A close analogy exists between this state of affairs and the recoil action in a gun: there the entire working pressure of the powder gases (barring friction) acts to accelerate the projectile, and reacts upon the gun and its mount with a force exactly equal to the inertia of the projectile; similarly the engine, accelerating the slide in one direction, is pushed the other way by a force identical with the inertia force of the slide. This force, rapidly changing and reversing, tends to produce a shaking effect — which will be a mere tremor, perhaps imperceptible, with massive foundations, but may become a very serious vibration where the foundation mass is relatively small, as in marine engines especially.

The two vertical forces, Q_G at W and at O, Fig. 156, form a couple;

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and since this is the only turning moment exerted upon the bed, it must, on general principles, be equal to, and in a sense the reaction against, the moment TR produced by the steam pressure upon the crank. It is through this couple that the load torque is felt by the engine bed.

If the load force, like L in Fig. 155, is steady and uniform, it simply develops a stress in the foundation bolts, which can easily be provided for.

(*g*) COUNTERBALANCING. — The device used to diminish the shaking effect of the inertia of the moving parts is shown in Fig. 157. This consists in placing an eccentric mass on the crank disc, opposite the

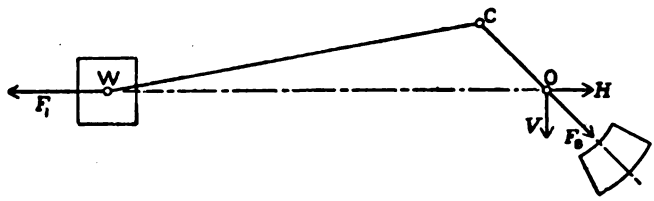

FIG. 157. — Effect of the Counterbalance.

pin: resolving its centrifugal force F_B into horizontal components H and V, we see that the former acts against F_1, reducing the free horizontal force to $(F_1 - H)$. Of course, this introduces a free, variant vertical force V; and the proper relative magnitude of $(F_1 - H)$ and V is a question to be determined by the conditions of service. The forces F_1 and H vary in about the same manner, as the crank rotates, so that the ratio between them remains nearly constant; and V varies like H, but with a different timing of cycle. The general idea in counterbalancing is to go as far as possible toward producing equilibrium within the system of inertia forces or of moving bodies. A discussing of simple counterweighting (with essentially the arrangement outlined in Fig. 157) is of most importance for the common two-cylinder locomotive, because the vertical component V enters into the pressure of wheel on rail, giving to this pressure a cyclical fluctuation. The more complex multiple-unit engines, with a number of cylinders and cranks, can be and generally are so arranged that the several sets of moving parts pretty effectively balance each other as to inertia effect, without the use of extra masses for counterbalance.

(*h*) ORDER OF PROCEDURE. — We have now covered in a general way the whole matter of the actions of the forces in the working mechanism or "main train" of the engine. The next step is to go over the same ground in detail, developing methods of determining the values of

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all the forces, and studying the manner and the effects of their variation. The steam pressure is determined (and determinable) only by the indicator diagram, whose form we already know: this leaves the inertia force as the important unknown quantity; and to find out its value we must make a complete study of the motion of the engine.

§ 31. Motion of the Engine Mechanism

(*a*) CONSTRAINED MOTION. — In any problem upon the motion of freely-moving bodies, it is necessary first to know the forces acting, and then to determine the resulting movements. In a machine, however, the parts can travel only in certain definite paths; and when, further, a major condition can be imposed which will entirely determine the motion of one part, those of the others may be derived by purely kinematic (that is, geometric) methods. In the steam engine this major condition is that the shaft or crank shall rotate at uniform speed.

This uniform rotation is, of course, secured by the use of a fly wheel, and in no engine is the uniformity absolute. But in the vast majority of cases the variations in rotary speed, within the revolution, are insignificant, and their effect upon the very large accelerations of the reciprocating parts is negligible.

(*b*) HARMONIC MOTION. — Before taking up the actual engine mechanism, consisting of bed, slide, connecting rod, and crank, we will discuss the simpler mechanism outlined in Fig. 158. Instead of the connecting rod, there is a cross-slot SS formed in the slide TT, in which

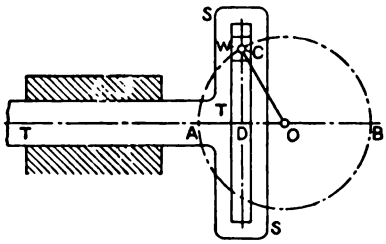

FIG. 158. — The Crossed Slider Crank.

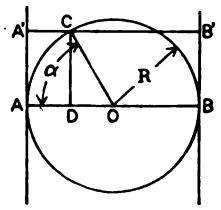

FIG. 159. — Diagram of Piston Displacement.

works the small block W surrounding the crank pin C. The simplest motion that can be gotten from, or determined by, a rotating crank is the harmonic motion of the projection D, of the pin center C upon the diameter AB, along that diameter. In the common mechanism this simple, symmetrical movement is modified by the effect of the angular swing of the connecting rod. In many approximate discussions, the ideal harmonic motion is used instead of the actual, because it is simpler,

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and because it represents the average working of the engine in both strokes.

Throughout the following discussions, these symbols will have invariable meanings:

R = radius of crank, or length of crank arm.
S = length of stroke, twice R.
L = length of connecting rod.
α = crank angle, measured from the left-side dead-center line OA, in the direction of rotation.
β = rod angle, between connecting rod and stroke line; see Fig. 159 or 162.
N = r.p.m. of shaft.
θ = angular velocity of crank.
s = piston travel or displacement, AD, from beginning of stroke.
v_0 = linear, tangential velocity of crank pin, taken to be constant in any particular case.
v = velocity of piston (or slide).
a_0 = acceleration of crank-pin center.
a = acceleration of slide.

The piston displacement is given by the diagram in Fig. 159: it is measured from A, and its value is

$$\text{AD} = \text{AO} - \text{DO}, \quad \text{or} \quad s = R(1 - \cos \alpha). \quad . \quad . \quad (151)$$

For the return stroke, from B to A, made while the crank is passing through the lower half of the circle, we can get the same relations by estimating α from OB and s from B; but in many cases it is better to carry α all the way from 0 deg. to 360 deg., keeping the same initial point for s.

Now it is self-evident that if we resolve either the velocity or the acceleration of C into vertical and horizontal components, the first will

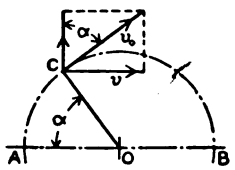

Fig. 160. — Velocity.

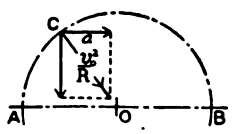

Fig. 161. — Acceleration.

belong to the motion of the block W in the slot, the second to the motion of the whole slide. Then in Fig. 160, v_0 being laid off from C perpendicular to OC, its horizontal component is

$$v = v_0 \sin \alpha. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (152)$$

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We know that when a point travels in a circular path with the velocity v_0, its acceleration, radially inward, is $a_0 = v_0^2/R$, where v is in feet per second and R in feet, if a_0 is to be in feet per second per second, the same term as g, the acceleration of gravity. Laying off this a_0 as shown in Fig. 161 and resolving, we get

$$a = a_0 \cos \alpha = \frac{v_0^2}{R} \cos \alpha. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (153)$$

(*c*) ANALYTICAL DERIVATION. — The same results can be obtained by an analytical method which will be especially useful when we come to the actual engine mechanism. In a very short time dt the piston will travel the distance ds; and, under the action of acceleration, the velocity will change by the amount dv: then from the primary definitions of velocity and acceleration,

$$v = \frac{ds}{dt}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (154)$$

$$a = \frac{dv}{dt} = \frac{d^2s}{dt^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (155)$$

Also, the relation between the linear motion of a point at the end of radius R and the angular motion of this radius is,

$$v_0 dt = R d\alpha;$$

both sides of the equation giving the distance traversed by the point C in the time dt; therefore

$$\frac{d\alpha}{dt} = \theta = \frac{v_0}{R} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (156).$$

Now starting with Eq. (151), $s = R(1 - \cos \alpha)$, we get by successive differentiation,

$$v = \frac{ds}{dt} = R \sin \alpha \frac{d\alpha}{dt} = v_0 \sin \alpha, \quad . \quad . \quad . \quad . \quad . \quad (157)$$

and

$$a = \frac{d^2s}{dt^2} = v_0 \cos \alpha \frac{d\alpha}{dt} = \frac{v_0^2}{R} \cos \alpha. \quad . \quad . \quad . \quad . \quad (158)$$

EXAMPLE 35. — For two typical cases, first of 48 in. stroke and 80 r.p.m., second of 15 in. stroke and 250 r.p.m., calculate angular velocity of crank, linear velocity of crank pin, and centripetal acceleration of crank pin. Also, assuming infinite connecting rod, find velocity and acceleration of the slide when the crank is at 60°.

These two engines have nearly the same piston speed (average), the numbers being 625 and 640 feet per minute, respectively.

The angular velocity θ is measured in radians per second, one revolution

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equaling 2π or 6.2832 radians. Running parallel columns for the two cases, the first results are,

Case 1.	Case 2.
$\theta = 2\pi\frac{N}{60} = 6.2832 \times 1.333 = 8.378$	$\theta = 6.2832 \times 4.167 = 26.180.$

Having θ computed, it is easiest to get the velocity, in feet per second, by $v_0 = R\theta$, with R in feet: otherwise, it is better to find circumference of crank circle in feet and multiply by revolutions per second. Using θ and reducing $\frac{1}{2}S$ to feet for R,

$$v_0 = \frac{S}{24} \times \theta = 2 \times 8.378 = 16.756 \qquad v_0 = 0.625 \times 26.180 = 16.363.$$

By circumference and revolutions per second the computation is,

$$v_0 = \frac{\pi S}{12}\frac{N}{60} = 12.566 \times 1.333 = 16.755 \qquad v_0 = 3.927 \times 4.167 = 16.363$$

For the centripetal acceleration, expressed in feet per second of velocity-change per second,

$$a_0 = \frac{v_0^2}{R} = \frac{16.756^2}{2} \quad \begin{array}{r} 1.22417 \\ 2.44834 \end{array} \qquad a_0 = \frac{16.363^2}{0.625} \quad \begin{array}{r} 1.21386 \\ 2.42772 \end{array}$$

$$= 150.4 \quad \begin{array}{r} 0.30103 \\ 2.14731 \end{array} \qquad = 428.4 \quad \begin{array}{r} 9.79588 \\ 2.63184 \end{array}$$

Note how much greater is the acceleration in the small "high-speed" engine, although both have nearly the same velocity of crank pin: the ratio of centrifugal force to gravity force, at the center of the crank pin, or $a_0 \div g$, is about 4.7 in the larger and 13.3 in the smaller engine.

Finally, with the crank at 60°, where $\sin\alpha = 0.8660$ and $\cos\alpha = 0.500$, the piston velocity and acceleration are,

$$v = v_0 \sin\alpha = 16.76 \times 0.866 = 14.53 \qquad v = 16.36 \times 0.866 = 14.17$$
$$a = a_0 \cos\alpha = 150.4 \times 0.5 = 75.2 \qquad a = 428.4 \times 0.5 = 214.2$$

(*d*) PISTON MOVEMENT. — The actual engine mechanism, sometimes called the slider-crank mechanism, is reduced to skeleton outline

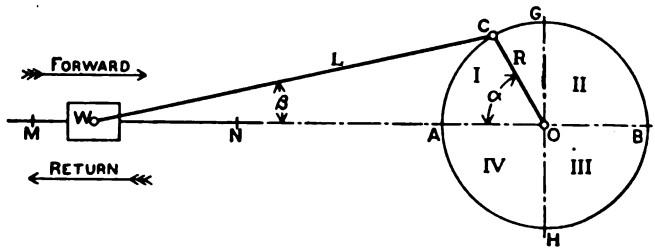

FIG. 162. — The Slider-crank Mechanism.

in Fig. 162. Crank travel is estimated from OA, all the way round the circle, the latter being divided into quadrants as indicated, and the

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piston movement is divided into forward and return strokes. When the crank is at A and the piston near the plain cylinder head, the engine is said to be on its head-end dead center; the other end, OB, is called the crank end. The whole piston slide is represented by the plain block at W.

The limits of the piston stroke are determined by making AM = BN = L. The piston travel, as shown in Fig. 163, is not AD, but MW, found by striking off CW from C as a center. A construction which will give this travel s for any crank position, without the trouble of thus striking off the rod length each time, is made by drawing the arcs A′A, B′B, tangent to the crank circle, from M and N as respective centers. Then for any position of C, it is only necessary to draw A′CB′ parallel to AB, and C will be located on this line just as W is on MN; for since the equal lines MA′, WC, and NB′ are included between parallels, they must be parallel; whence A′C = MW.

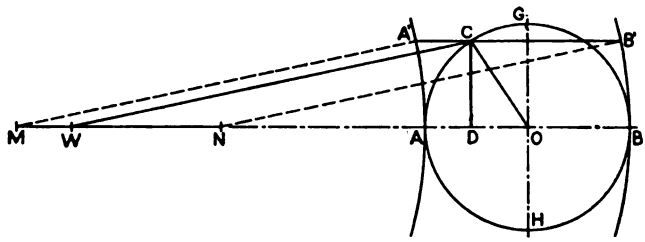

FIG. 163. — Piston Travel.

The analogous construction in Fig. 159 is made by drawing the straight-line tangents A′A, B′B: and it appears that the effect of the connecting rod is to introduce a curvature into these limit lines. Since the straight lines in Fig. 159 may be thought of as arcs of infinite radius, the motion there represented is often spoken of as that with infinite connecting rod. Another view of the same conception is, that if the rod were of infinite length its angular swing would be zero, and the piston would receive harmonic motion without distortion. In Fig. 158, this "infinite rod" is replaced by the simple slide block W, the piece which forms the connection between the crank pin and the slide; and to carry out the idea still farther, we may think of the surface of the slot as a small part of the surface of a wrist pin of infinite radius.

EXAMPLE 36. — With a connecting rod five cranks in length, how far is the piston from mid-stroke when the crank is vertical, or at 90° in Fig. 163?

The mid-stroke position of W is at the distance $L = nR$ to the left of 0. When C is at G, or $\alpha = 90°$, OW will be the base of a right triangle of which L

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is the hypotenuse and R the other side, so that $OW = \sqrt{L^2 - R^2} = R\sqrt{n^2 - 1}$. The distance sought, expressed in terms of the stroke is then

$$s_0 = \frac{n - \sqrt{n^2 - 1}}{2} S. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (159)$$

Substituting $n = 5$, we get,

$$s_0 = \frac{5 - \sqrt{24}}{2} S = \frac{5 - 4.899}{2} S = 0.0505\, S.$$

(*e*) VELOCITY AND ACCELERATION. — The analytical method will be employed first, as it is shorter and gives results in a very useful form. From Fig. 163 we get for the travel s the expression

$$\begin{aligned} s &= MW = MO - WO \\ &= (L + R) - (L\cos\beta + R\cos\alpha) \\ &= R(1 - \cos\alpha) + L(1 - \cos\beta); \quad . \quad . \quad . \quad . \quad (160) \end{aligned}$$

where the second term, $L(1 - \cos\beta)$, shows the departure from the harmonic motion represented by Eq. (151), or the effect of the connecting rod. This is evidently the distance, on the line A′B′, between the curved limit lines AA′, BB′ of Fig. 163 and the respective corresponding straight lines of Fig. 159.

To eliminate the rod angle β, we note that, in Fig. 163,

$$CD = R\sin\alpha = L\sin\beta;$$

whence

$$\sin\beta = \frac{R}{L}\sin\alpha,$$

$$\cos\beta = \sqrt{1 - \sin^2\beta} = \left(1 - \frac{R^2}{L^2}\sin^2\alpha\right)^{\frac{1}{2}}.$$

If this is developed by the binomial formula, the first two terms are

$$\cos\beta = 1 - \frac{1}{2}\frac{R^2}{L^2}\sin^2\alpha + \text{* * *}; \quad . \quad . \quad . \quad (161)$$

and the succeeding terms, involving high powers of fractions, may be dropped without overpassing the limits of desired accuracy. Then Eq. (160) becomes

$$s = R\left(1 - \cos\alpha + \frac{1}{2}\frac{R}{L}\sin^2\alpha\right);$$

and from this

$$\begin{aligned} v = \frac{ds}{dt} &= R\left(\sin\alpha + \frac{1}{2}\frac{R}{L}\sin 2\alpha\right)\frac{d\alpha}{dt} \\ &= v_0\left(\sin\alpha + \frac{1}{2}\frac{R}{L}\sin 2\alpha\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad (162) \end{aligned}$$

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Likewise

$$a = \frac{dv}{dt} = v_0\left(\cos\alpha + \frac{R}{L}\cos 2\alpha\right)\frac{d\alpha}{dt}$$
$$= \frac{v_0^2}{R}\left(\cos\alpha + \frac{R}{L}\cos 2\alpha\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (163)$$

A strictly exact formula for a in terms of α can be worked out, avoiding the approximation of Eq. (161); but it is quite complex, and the gain in accuracy is of no practical significance.

By purely graphical methods, relations which parallel Eqs. (162) and (163) may be derived: the first, for piston velocity, will now be established, because it is useful in connection with the matter of turning force on the crank; but that for acceleration is lengthy in deduction and decidedly less convenient in application than constructions based upon Eq. (163).

EXAMPLE 37. — For a given rod ratio $n = L/R$, find the crank angle at which the piston velocity is maximum.

Velocity v will be at its maximum when acceleration a is zero. Solving the equation

$$\cos\alpha + \frac{1}{n}\cos 2\alpha = 0,$$

we get

$$n\cos\alpha + 2\cos^2\alpha - 1 = 0,$$
$$\cos^2\alpha + \tfrac{1}{2}n\cos\alpha = \tfrac{1}{2}:$$

whence

$$\cos\alpha = \tfrac{1}{4}(\sqrt{8+n^2} - n). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (164)$$

If, for instance, n is 5, $\cos\alpha$ comes out 0.1862 and α is 79° 16′. The sine of this angle is 0.9825 and that of 2α or 158° 32′ is 0.3660. Substituting in Eq. (162), we have

$$v = (0.9825 + 0.0366)\, v_0 = 1.0191\, v_0.$$

(*f*) VELOCITY BY GRAPHICAL RELATIONS. — It is a fundamental principle of kinematics, that if a body has motion in a plane, and if the directions of movement of two points of the body are known, the intersection of lines perpendicular to these motion directions determines the instantaneous center of rotation — a point about which, *at the instant*, the body is rotating as about a fixed pivot. In Fig. 164, CP and WP are drawn perpendicular to the paths (or to the velocity directions) of C and W, and P is the instantaneous center.

If now the connecting rod can be thought of as turning, for the instant, about P, the conclusion follows that the velocity of any point on the rod is perpendicular to a radius from P to this point, and is proportional to the length of this radius. The fact that P changes its position from instant to instant, or from position to position of the mechanism,

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traveling along a curved path or locus, does not at all invalidate the statement just made as to the velocity relations at a particular instant.

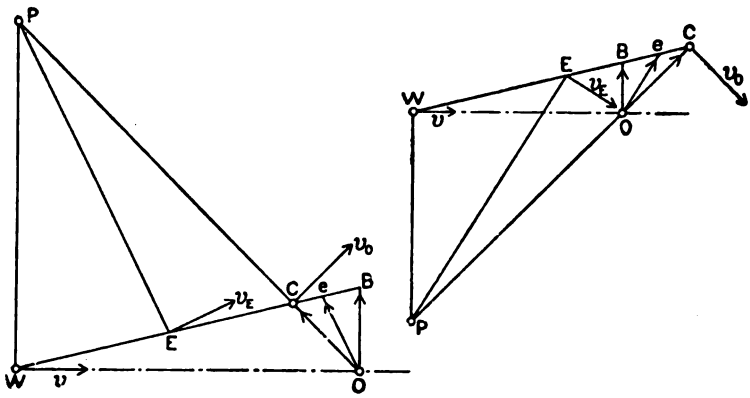

FIG. 164. — The Instantaneous Center of Rotation.

For the point W only the second of these two relations need be used, as the direction of v was part of the data: and we have

$$v : v_0 : : \text{PW} : \text{PC}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (165)$$

For any other point as E, we first draw a direction line for v_E, and then find its value by

$$v_E : v_0 : : \text{PE} : \text{PC}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (166)$$

A practical disadvantage of this method is that it involves a lot of troublesome graphical work, in the drawing of long lines to get P, and in satisfying the proportion (165) or (166). A much more convenient

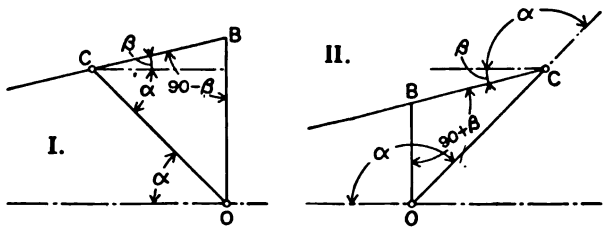

FIG. 165. — Velocity Relations.

construction suggests itself when we note that the line OB, through the shaft center O, perpendicular to the stroke line WO, and meeting the rod line WC at B, makes a triangle OBC similar to PWC, so that

$$v : v_0 : : \text{OB} : \text{OC}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (167)$$

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And if, further, such a scale be chosen that $OC = v_0$, then at once $OB = v$.

The trigonometrical relation between OB and OC is illustrated in Fig. 165, which shows parts of Fig. 164 enlarged. In the triangle OBC,

$$\frac{OB}{OC} = \frac{\sin OCB}{\sin OBC} = \frac{\sin(\alpha + \beta)}{\cos \beta} = \frac{v}{v_0} \quad . \quad . \quad . \quad . \quad . \quad (168)$$

(*g*) DIAGRAM OF PISTON VELOCITY. — In Figs. 164 and 165 are shown the two typical conditions, for the first and second quadrants, respectively, of the crank motion. In quadrant I, OB/OC is greater than sin α, in quadrant II it is less: while for infinite rod, CB would be horizontal and the motion symmetrical with respect to the vertical line OB. It is not necessary to make a separate determination for the lower half of the crank circle, because all kinematic and geometrical relations are symmetrical above and below the stroke line.

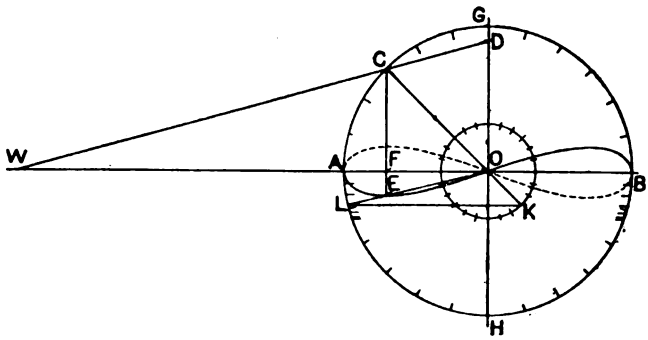

FIG. 166. — Derivation and Form of the Diagram of Piston Velocity.

In Fig. 166, the radius OC is taken to represent, on a convenient scale, the intensity of the (constant) velocity v_0 of the crank pin; then, directly from Fig. 165, the rod line WC cuts off, in OD, the length of the piston velocity. It is troublesome to be compelled to draw the extended mechanism outline OCW in order to get the slant of the rod: a far more convenient method is the miniature construction KOL. The full radius OC or OL is taken to represent the rod, the short radius OK the crank: thus if the rod is six cranks long, OK will be one-sixth of OL. In learning to use this device, it is enough to keep in mind the general slant of the rod CW; a clear sense of what is wanted is a better guide than a formulated rule in the selection of the intersections K and L.

Instead of drawing CD parallel to LO, drop a vertical line CE (perpendicular to the stroke line); then the intercept CE, equal to DO,

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will be the velocity v, while CF is the similar quantity for the case of harmonic motion. The full-line curve AEOB is the locus of the inner end of the ordinate from C for the forward stroke, or for crank arc AGB, the dotted curve BOA the corresponding locus for the return stroke, arc BHA. The diagram shows very clearly how the connecting rod causes the piston to move relatively faster in the first and fourth quadrants, slower in the second and third.

(*h*) ACCELERATION OF THE SLIDE. — The similar diagram for acceleration is based on Eq. (163). Evaluating the factor

$$m = \left(\cos\alpha + \frac{R}{L}\cos 2\alpha\right),$$

we have

$\alpha =$	$0°$	$90°$	$180°$	$270°$
$m =$	$\left(1+\frac{R}{L}\right)$	$-\frac{R}{L}$	$-\left(1-\frac{R}{L}\right)$	$-\frac{R}{L}$

Also, for $\alpha = 45$ deg., 135 deg., etc. (that is, for any mid-quadrant crank position) $m = \pm\cos\alpha$.

Now in Fig. 167, the radius of the circle is a_0: and for infinite rod it is evident that the horizontal distance CE′, from the C-point to the line GH, will give a, according to the relation $a = a_0\cos\alpha$. For the actual mechanism, GH must be replaced by a curve analogous to the D-curve in Fig. 166: three points in this curve are got by laying off, from m as above,

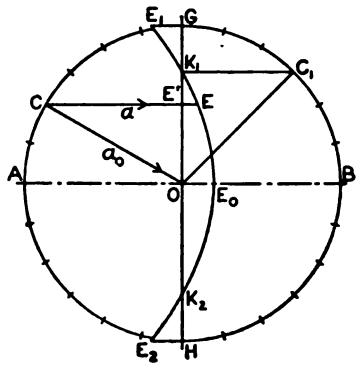

FIG. 167. — Acceleration Diagram.

$$OE_0 = GE_1 = HE_2 = \frac{R}{L}AO;$$

and two more by noting that the curve must cross GH where $m = \cos\alpha$, at the mid-quadrant points K_1 and K_2. Without any further determinations (unless the figure be very large), a fair curve can be drawn through these five points which will be the locus of the end of a, measured always from C toward the E-curve.

As indicated in the laying out of points E_1, E_0, and E_2, an acceleration a is considered plus when measured from C to E toward the right in Fig. 167, and then acts toward the right. In Fig. 166, an ordinate from C down to E is positive, showing a velocity toward the right: the opposite direction corresponds with the minus sign in both cases. For

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crank positions which are symmetrical with reference to AB, the accelerations are absolutely the same, as for 60 deg. and 300 deg.; but referred to the velocities existing at such corresponding positions, these accelerations are relatively opposite.

§ 32. Working Forces in the Engine

(*a*) INERTIA FORCE OF THE RECIPROCATING PARTS. — Knowing the acceleration a and the weight W of the sliding parts — which are supposed to include a portion of the connecting rod — we have only to multiply the mass M (equal to $W \div g$) by a in order to get the force $F = Ma$ required to accelerate this mass, or its inertia force. Then for the mechanisms discussed we have,

For infinite rod,

$$F = \frac{W}{g}\frac{v_0^2}{R}\cos\alpha; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (169)$$

For actual rod,

$$F = \frac{W}{g}\frac{v_0^2}{R}\left(\cos\alpha + \frac{R}{L}\cos 2\alpha\right). \quad . \quad . \quad . \quad . \quad . \quad (170)$$

In either case, $\frac{W}{g}\frac{v_0^2}{R}$ is the centrifugal force which the reciprocating mass would have if it were concentrated around the crank-pin center C: this ideal centrifugal force will be called F_0, and the actual inertia force is then a component of F_0, given by

$$F = F_0 \cos\alpha, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (171)$$

or by

$$F = F_0\left(\cos\alpha + \frac{R}{L}\cos 2\alpha\right). \quad . \quad . \quad . \quad . \quad . \quad (172)$$

The value of F can be found, for any crank angle, by a diagram like Fig. 167, but with F_0 instead of a_0 as the radius. As to direction

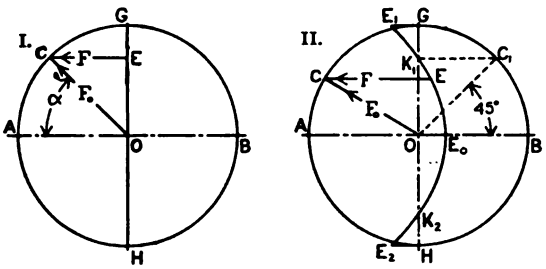

FIG. 168. — Circular Diagrams of Inertia Force.

of force, it must be remembered that accelerating force will point toward the center O or toward the limit line GH or $E_1E_0E_2$; while

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inertia force will point outward, or away from the middle of the diagram. This is illustrated in Fig. 168, where the two cases as to form of mechanism are separated, the two diagrams embodying Eqs. (171) and (172), respectively.

These diagrams are not, however, in a shape suitable for the direct combination of inertia force with steam pressure, since the latter, given by the steam diagram, is laid out on a stroke-line base The derivation of an inertia diagram in these terms is shown in Fig. 169, where, in I,

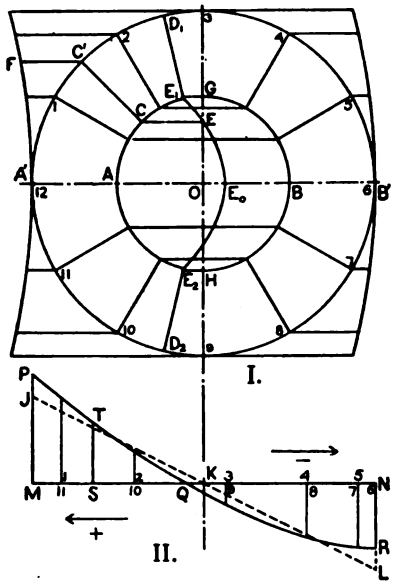

Fig. 169. — The Stroke-line Diagram of Inertia Force.

the diagram from Fig. 168 II is surrounded by another diagram, for piston position, like Fig. 163. From this we get the two coördinates for the curve in II, the abscissa s = MS at FC′, the ordinate F = ST at CE. For the actual mechanism this gives the curve PQR, for infinite rod, the straight line JKL. It must be clearly understood that this is a diagram for both strokes, as is indicated by the numbered positions on both figures. The meaning of plus and minus (up and down) ordinates in II is shown by the arrows.

That JL must be a straight line is evident when we consider that, with infinite rod, the distance SK from mid-stroke and the inertia force F are both proportional to cos α. With the curve, the areas PMQ, QNR — the first representing work stored in the moving parts from zero velocity to maximum, the second that given back during retarda-

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tion — must be equal. Note that the dead-center acceleration is always greater than a_0 at the head end and less at the crank end, the ends being taken according to the conventional arrangement in Fig. 162.

As written Eqs. (171) and (172) give the total inertia force in the engine. For combination with steam-pressure diagrams it is usually desirable to have the force per square inch of piston, so that F_0/A and F/A are more commonly dealt with than F_0 and F.

As regards the division of the mass of the connecting rod, for "concentration" at the pins — see § 30 (*c*) — to put one-half at each center serves very well when the driving force is in question. It is more logical, because strictly correct when the result sought is the total "shaking force" of § 30 (*f*), to use the division got by weighing the two ends on knife edges at or under the axes of the pin bearings. This keeps the center of gravity of the rod unchanged: thus if the center is at 0.6 of the distance from wrist pin W to crank pin C, 0.6 of the mass goes to C, 0.4 to W.

EXAMPLE 38. — A 14 in. by 15 in. engine, which runs at 250 r.p.m., has a connecting rod 42 in. long, and the total reciprocating weight is 325 lb. Find F_0 and F_0/A, also the value of F/A at the dead centers and at 50° and 125° of crank angle.

From Example 35, the centripetal acceleration a_0 of the crank-pin center is 428.4 ft. per sec. per sec. Then the ideal centrifugal force is

$$F_0 = \frac{W}{g} a_0 = \frac{325 \times 428.4}{32.16} = 4329 \text{ lb.}$$

The area of the 14 in. piston is $A = 153.9$ sq. in., so that

$$\frac{F_0}{A} = 4329 \div 153.9 = 28.12 \text{ lb. per sq. in.}$$

Here $\frac{R}{L} = \frac{7.5}{42} = \frac{1}{5.6}$; then at head-end dead center,

$$\frac{F}{A} = 28.12 + \frac{28.12}{5.6} = 28.12 + 5.02 = 33.14 \text{ lb.;}$$

and at the crank-end dead center,

$$\frac{F}{A} = 28.12 - 5.02 = 23.10 \text{ lb.}$$

Now evaluating $m = \left(\cos\alpha + \frac{R}{L}\cos 2\alpha\right)$, we have at 50°,

$$m = 0.6428 - \frac{0.1737}{5.6} = 0.6428 - 0.0310 = 0.6118;$$

and at 125°,

$$m = -0.5736 - \frac{0.3420}{5.6} = -0.5736 - 0.0611 = -0.6347.$$

Then the required values of F/A are, 17.20 lb. at 50° and −17.85 lb. at 125°.

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(*b*) Effective Driving Force. — In Fig. 170 a pair of indicator diagrams, from the two ends of the cylinder, is shown at I: and by means of motion arrows the fact is made clear that the forward-pressure line, or steam line of one end, as AB, is simultaneous with the back-pressure line, or exhaust line of the other end, as GH. Then a subtraction of P_B from P_F for the whole of each stroke, giving P_E — refer to Fig. 153 — is made by the combination in II; where MGHN (crank end) is superimposed on MABN (head end), and where the effective

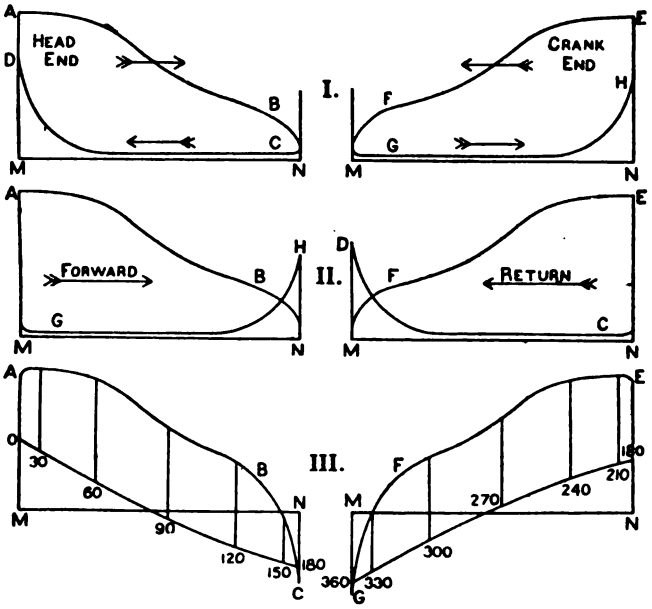

Fig. 170. — Effective Driving Force.

steam pressure P_E is given by the ordinate intercepted between the curves AB and GH. Note that whereas the indicator cards in I are for the two ends of the cylinder, the two diagrams in II are for the respective strokes (compare § 21 (*b*): and further that, with high compression, the back pressure will rise above the forward pressure toward the end of each stroke, so that the mechanism will have to drag the piston to dead center instead of being driven by it. In § 30 (*b*), the pressures P_F and P_B were defined as absolute pressures, above perfect vacuum; as shown in Fig. 130, on diagrams from a noncondensing engine, they are measured above atmosphere: but since their difference is the result sought, either datum line may be used.

The diagrams in Fig. 170 II, where the variable ordinate p_x is included, between curves, are brought to a more convenient form in

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III, by measuring the ordinate up (and down) from MN, and getting the p_x-diagrams MABCN, NEFGM, on straight base lines. The inertia-force diagram can be combined directly with these by laying it off on the same base line — being inverted for the return stroke because the direction of positive or forward-acting steam pressure is there reversed. This shows clearly how the inertia force diminishes the effective pressure during the first part of each stroke and increases it toward the end — an action which, with the usual form of indicator diagram, greatly favors a uniform distribution of driving force and of transmitted work throughout the stroke: in this particular case, the negative pressure toward the end of the stroke, without inertia, is almost entirely eliminated when the latter is taken into account.

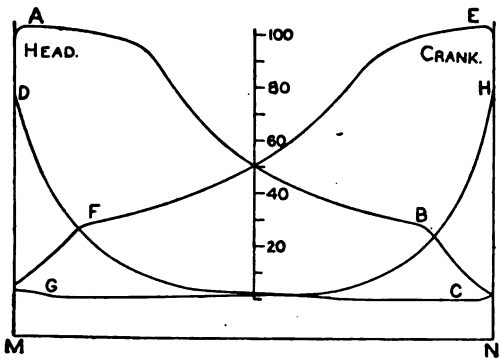

FIG. 171. — Indicator Diagrams from Fig. 84.

(c) THE DOUBLE DIAGRAM OF DRIVING FORCE. — To illustrate another method of combination and to serve as a basis for some further determinations, the diagrams in Fig. 84 are brought to a common base line in Fig. 171, reversed into the standard position and with irregularities due to the indicator smoothed out. Effective steam pressures can now be measured directly from this double indicator diagram, between curves AB and GH for the forward stroke, between EF and CD for the return stroke. These intercepts are laid off from the straight-line base in Fig. 172, forces acting toward the right being measured upward and those toward the left downward. Since now there is only one direction meaning of the ordinates, a single inertia curve serves for both strokes: to get effective driving force at wrist pin, measure from PQR to ABC for the forward stroke and to DEF for the return stroke.

In a vertical engine, the weight of the reciprocating parts acts as a constant force in the direction of their movement, increasing driving force in the down stroke, diminishing it in the up stroke — the cylinder

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being, of course, above the shaft. In Figs. 170 III and 172, this can easily be taken into account by a simple shifting of the base line MN (and with it the inertia curve) through a distance equal to W/A.

It is apparent that the inertia-force action of these moving masses tends to equalize the distribution of driving work, storing up the excess in the first part of each stroke and delivering it in the latter part, when the effective steam pressure has fallen low or even reversed in direction.

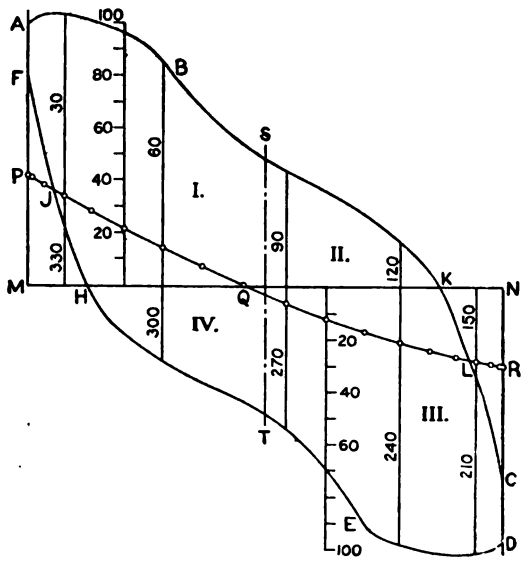

FIG. 172. — The Two-stroke Diagram.

(*d*) TURNING-FORCE RELATIONS. — Two methods of reasoning may be followed in finding, for a known driving force at the wrist pin, the turning effect upon the crank: both are under the assumption that the connecting rod is a weightless transmitting link, its inertia having been taken into account according to § 30 (*c*). The first is illustrated in Fig. 173, where the force S transmitted along the rod is the resultant of the driving force P and the guide reaction Q. The perpendicular OE from O upon the rod line is the lever arm of this force, and its moment is

$$M = S \times \text{OE}.$$

Drawing also the vertical line OB, we have that the triangles OEB, WFH, are similar: wherefore

$$P : S : : \text{OE} : \text{OB},$$

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or

$$P \times \text{OB} = S \times \text{OE} = M: \quad . \quad . \quad . \quad . \quad . \quad . \quad (173)$$

and the turning effect is just as though the force P acted upon the crank with the lever arm OB.

In the mechanism of Fig. 158, or with "infinite connecting rod," the force P would be applied horizontally, or parallel to the stroke line, at C, and its lever arm would be CD or $R \sin \alpha$: the increase of arm length from CD to OB measures the effect of the connecting rod. In the second quadrant, OB will be less than CD.

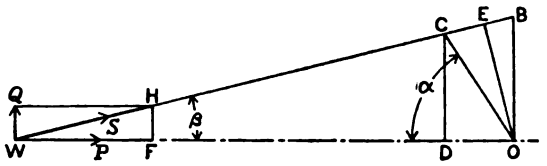

FIG. 173. — Turning-moment on the Crank.

Replacing the moment M in Eq. (173) by that of a tangential force T at C, with the arm R, we have

$$M = P \times \text{OB} = T \times R,$$

or

$$T = \frac{\text{OB}}{R} \times P = \frac{\sin(\alpha + \beta)}{\cos \beta} P; \quad . \quad . \quad . \quad . \quad . \quad (174)$$

the geometrical relation being the same as that for the velocities in Fig. 165, and Eq. (174) derived like Eq. (168). The relation

$$\frac{v}{v_0} = \frac{T}{P} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (175)$$

which comes from a combination of these two equations, is necessarily true from fundamental principles. Under the assumption of no frictional or internal losses in the machine, work rate at wrist pin must equal work rate at crank pin. This rate is equal to the product of working force by its velocity (along the force line); therefore the equation

$$vP = v_0 T \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (176)$$

must be true.

The second derivation of T is through force analysis strictly, and is shown in Fig. 174. Here the rod force S is carried over to C and there resolved, as suggested in Fig. 153. From triangle WFH or CED,

$$S = \frac{P}{\cos \beta};$$

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and in triangle CGD, right-angled at G, and with angle CDG = angle OCD = $(\alpha + \beta)$,

$$T = S \sin(\alpha + \beta) = P \frac{\sin(\alpha + \beta)}{\cos \beta} \quad . \quad . \quad . \quad . \quad . \quad (177)$$

If we apply to this expression the effect of making the rod infinite — which makes β zero — we get at once the simple relation indicated by the lever arm CD or $R \sin \alpha$ in Fig. 173, namely,

$$T' = P \sin \alpha; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (178)$$

which is, of course, the same as the velocity relation for that mechanism, in Eq. (152).

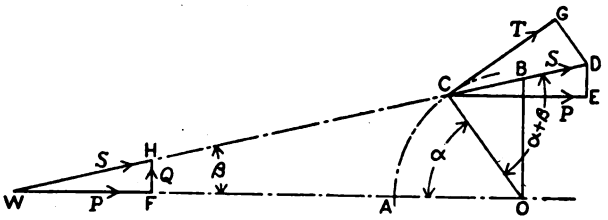

FIG. 174. — Tangential Force on Crank.

(e) DETERMINING TANGENTIAL FORCE. — Having drawn the diagrams of effective driving force, like Fig. 170 III or Fig. 172, and located on them a series of ordinates corresponding to a number of equally-spaced crank angles, the next step is to find the turning force T for each of these P's. One method, shown and used in Fig. 175, is a direct application of the relation expressed in Eq. 174), which is equivalent to

$$T : P : : \text{OB} : \text{OC} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (179)$$

on Fig. 174: and the construction is similar to that for v in Fig. 166. The crank circle on AB is of any convenient size, but with its radius greater than the largest value of P on the diagram of effective driving force. Each length of P, taken from Fig. 170, is measured inward along the corresponding crank line, as CD; and the length CE, cut from the vertical by DE parallel to the rod line, is T. The triangle CDE is similar to OCB in Figs. 173 and 174; and the construction in Fig. 166 is used for finding the rod angle. A separate triangle must be drawn for each crank position. The T's, when found, are laid off radially outward from the circle, and a curve is traced through their ends.

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TABLE 15. TURNING-FORCE RATIOS.

Values of $m = \frac{\sin(\alpha+\beta)}{\cos\beta} = \frac{T}{P} = \frac{v}{v_0}$, for different values of $n = \frac{L}{R}$.

α°		Values of n.				
		4	5	6	8	∞
5	355	.1089	.1045	.1016	.0980	.0872
10	350	.2164	.2079	.2022	.1950	.1737
15	345	.3215	.3089	.3005	.2901	.2588
20	340	.4227	.4065	.3957	.3822	.3420
25	335	.5189	.4995	.4866	.4706	.4226
30	330	.6091	.5870	.5724	.5542	.5000
35	325	.6923	.6682	.6523	.6325	.5736
40	320	.7675	.7421	.7253	.7054	.6428
45	315	.8341	.8081	.7910	.7699	.7071
50	310	.8915	.8657	.8488	.8279	.7660
55	305	.9392	.9144	.8982	.8782	.8192
60	300	.9769	.9540	.9390	.9205	.8660
65	295	1.0046	.9842	.9709	.9545	.9063
70	290	1.0224	1.0051	.9939	.9801	.9397
75	285	1.0303	1.0169	1.0082	.9974	.9659
80	280	1.0289	1.0197	1.0137	1.0064	.9848
85	275	1.0186	1.0139	1.0109	1.0072	.9962
90	270	1.0000	1.0000	1.0000	1.0000	1.0000
95	265	.9738	.9785	.9815	.9853	.9962
100	260	.9407	.9499	.9559	.9633	.9848
105	255	.9015	.9150	.9237	.9344	.9659
110	250	.8570	.8742	.8855	.8992	.9397
115	245	.8080	.8284	.8417	.8581	.9063
120	240	.7552	.7781	.7931	.8116	.8660
125	235	.6991	.7239	.7401	.7601	.8192
130	230	.6406	.6664	.6833	.7042	.7660
135	225	.5801	.6061	.6232	.6444	.7071
140	220	.5181	.5435	.5603	.5810	.6428
145	215	.4549	.4790	.4949	.5147	.5736
150	210	.3909	.4130	.4276	.4458	.5000
155	205	.3263	.3458	.3586	.3747	.4226
160	200	.2614	.2776	.2884	.3018	.3420
165	195	.1962	.2088	.2171	.2276	.2588
170	190	.1309	1394	.1451	.1523	.1737
175	185	.0654	.0698	.0727	.0763	.0872

A second method uses the computed values of the ratio of T to P, or of $m = \frac{\sin(\alpha+\beta)}{\cos\beta}$, given in Table 15, working through a reduction or proportion diagram like those in Fig. 81. The base line AB, Fig. 176, is taken of any convenient length, and along the perpendiculars AC

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and BD are measured values of $m \times$ AB. Note that m rises from zero to a little more than 1.00, then comes back to zero at the other dead center. To avoid overlapping of the ascending and descending

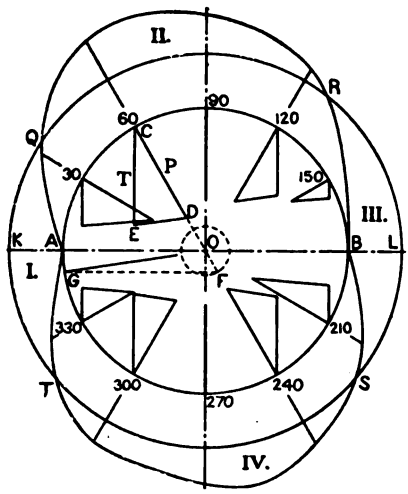

FIG. 175. — Turning-force Diagram.

series of lines, values belonging to quadrants I and IV are laid up on AC, those for quadrants II and III on BD. Then the inclined lines are drawn: and for any particular crank angle, as 20 deg., we have only

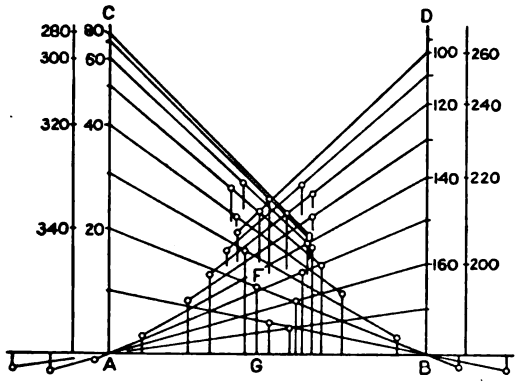

FIG. 176. — Transformation Diagram.

to measure off from B, as BG, the length of P taken from its diagram, in order to get the corresponding T in GF.

(*f*) DIAGRAMS OF TURNING FORCE. — Values of T having been de-

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termined, they can be laid out either from the crank circle, as in Fig. 175, or from a straight-line base, as in Fig. 177. This latter base line is the developed crank circle: that is, the upper and lower halves of the circular diagram are straightened out and brought together. The circular diagram is clearer for illustrating the continuous variation in turning force, while the other is better for quantitative determinations.

Assuming that the load on the engine is a uniform torque, and either that there are no frictional losses or that their effect is included in the load, we get the value of the equivalent resistance at the crank circle by finding the mean tangential force through Eq. (150). Measuring off this T_m at AK, we draw the resistance circle KL on Fig. 175: and by comparing the curve of actual turning force, AQRBST, with this circle of uniform resistance, we get an idea of the duty which the fly wheel has to perform. Only at four points, Q, R, S, and T, is the

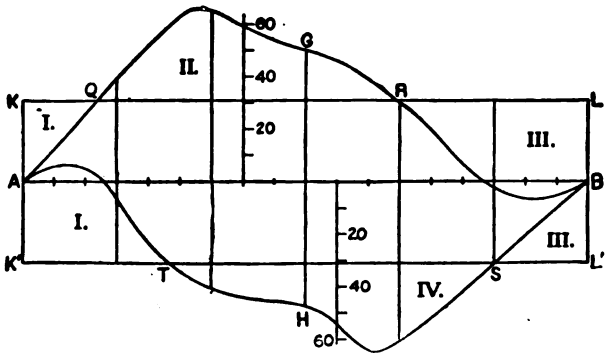

FIG. 177. — Diagram on Developed Circle.

driving force just equal to the resistance: during two periods or phases, marked I and III, the work done upon the shaft is less than that taken from it, and the deficiency is supplied by energy from the wheel, which of course slows down; but during phases II and IV the wheel has to store up excess work, and regains kinetic energy. Knowing the largest amount of work to be taken care of by the wheel, as shown by the diagram, and the greatest permissible variation of speed within the revolution, we can compute the weight of wheel rim required under any conditions.

(*g*) FLY-WHEEL DATA. — Both forms of the turning-force diagram, Figs. 175 and 177, show force plotted on a distance base: but only in the straight-base diagram, where the ordinates are parallel, is work truly represented by area. Working then on Fig. 177, we can get the data for fly-wheel determination in several forms.

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Measuring the area of each phase and dividing area in square inches by base in inches, we get the mean height and reduce it to the pressure scale: note that the bases of the broken phases I and III in Fig. 177 are KQ + K'T, RL + SL'. Besides this, the four phase lengths are to be expressed in degrees, so as to measure the angles TOQ, QOR, ROS, SOT in Fig. 175. Let

γ = this phase angle, in degrees;
l = length of phase, on crank circle, in feet;
t = average unbalanced force acting during phase, in pounds per square inch of piston, so that At is the total force for the engine;
E = work value of the phase, in foot-pounds.

The circumference of the crank circle is

$$C = \frac{\pi S}{12} \text{ ft.};$$

then
$$l = \frac{\gamma}{360} C \text{ ft.}, \quad \text{and} \quad E = tAl \text{ ft. lb.} \quad . \quad . \quad . \quad . \quad (180)$$

Again, in Fig. 177, the whole area of the diagram AGBH, or the equivalent rectangle KLL'K', represents the work of the engine in one revolution, which we may call W_R: and if we find the ratio k which each phase area bears to the total area, we can get E through the relation

$$E = kW_R. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (181)$$

If we know the average m.e.p. or p_m, we find W_R by

$$W_R = 2\frac{S}{12} A p_m. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (182)$$

Or, having the i.h.p. H and the r.p.m. N,

$$W_R = \frac{33{,}000\, H}{N}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (183)$$

Finally, knowing the dimensions of the engine and the scales used in laying out Fig. 177, we may calculate a work scale of foot-pounds per square inch, as follows:

Suppose that Fig. 177 is drawn for a 14 in. by 15 in. engine, and that in the figure the circumference scale is 1 in. = 20 deg., while that for pressures is 1 in. = 24 lb. per sq. in. Then the circumference of the 15 in. crank circle is 3.927 ft., and the distance value of 1 in. along the base of the diagram is

$$\frac{20}{360} \times 3.927 = 0.2182 \text{ ft.}$$

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The piston area being 153.9 sq. in., the total force value of 1 in. of ordinate is $24 \times 153.9 = 3694$ lb.: so that 1 sq. in. represents

$$0.2182 \times 3694 = 806.0 \text{ ft. lb.} \quad . \quad . \quad . \quad . \quad . \quad (184)$$

A table of data from Fig. 177, applying it to an engine 14 in. by 15 in. at 250 r.p.m., is shown below. With the degree scale 1 in. = 20 deg., the length of the diagram must be 9 in.: the measured area of the whole figure AGBH was 23.17 sq. in.: so that the mean turning force or m.t.f. is

$$T_m = \frac{23.17}{18} \times 24 = 30.9 \text{ lb.}$$

Applying Eq. (150) backward, the m.e.p. corresponding would be

$$P_m = 30.9 \div 0.6366 = 48.5 \text{ lb.},$$

as against an average value of 48.03 on page 164: the discrepancy is due to inaccuracy in measurement and transference of the small indicator diagrams.

TABLE 16. RESULTS FROM FIG. 177.

Phase No.	Area, sq. in. a	Angle, degrees. γ	Length, feet. l	Mean force. t	Phase-ratio. k	Work-value. E
I.	−3.34	70.6	0.770	−23.4	.1442	−2690
II.	+3.92	95.0	1.036	+19.4	.1715	+3160
III.	−4.23	91.0	0.993	−22.3	.1824	−3410
IV.	+3.67	103.4	1.129	+17.0	.1584	+2960

In Table 16, the phase areas a were measured with the planimeter, as was that of the whole figure: given as measured, they check by adding up very nearly to zero. Dividing each by its base in inches, and multiplying by the force scale 24, gives the mean force t. The phase angles γ were measured with a scale of 20 to the inch and made to add up to 360 deg.: and the corresponding lengths l on the crank circle were got by multiplying γ by the value of 1 deg. in feet, which is $3.927 \div 360 = 0.01091$ ft. The phase area a divided by the total area 23.17 sq. in. gives the ratio k. Finally, E is got from a through the work scale of Eq. (184).

The work per revolution, W_R, as found from the total area 23.17 sq. in., is

$$W_R = 23.17 \times 806 = 18{,}650 \text{ ft. lb.}$$

By Eq. (182) it is

$$W_R = \frac{30}{12} \times 153.9 \times 48.5 = 18{,}650 \text{ ft. lb.}$$

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The supply of data in this table is redundant, more being given than is necessary for the solution of any particular problem. It will be noted that a, γ, t, and k depend only upon the form of the diagram, while l and E involve also the dimensions of the engine.

§ 33. Fly-wheel Action

(a) Weight of Wheel. — The variation work E, just determined, is to be taken up or given off by the wheel within a certain prescribed limit of speed change. This limit is generally defined by stating that the range from the greatest velocity V_1 to the least velocity V_2 of the wheel rim is not to exceed a certain fraction f of the average velocity V. The change in kinetic energy of a mass whose weight is W, in a drop of speed from V_1 to V_2 (or the reverse), is

$$E = \frac{W}{g}\frac{(V_1^2 - V_2^2)}{2} = \frac{W}{g}\frac{(V_1 + V_2)}{2}(V_1 - V_2).$$
$$= \frac{W}{g} \times V \times fV = f\frac{WV^2}{g} \quad \ldots\ldots\ldots \quad (185)$$

This E is identical with that in Eqs. (180) and (181): so that the method of solving the fly-wheel problem consists in getting from the diagram an expression for, or value of, the work to be taken care of by the wheel, and equating this to the expression for change in kinetic energy.

The fraction f of permitted fluctuation varies with the character of the work done by the engine: a good degree of uniformity, suitable for ordinary high-grade work such as driving electric generators and textile mills, is secured by making f = about 0.01. In slow-running engines on rough work, the fluctuation often greatly exceeds this value, rising to 5 or 10 per cent or more; while in the work requiring the greatest delicacy of regulation — the driving of alternating-current generators in parallel — the fly wheel must be very powerful.

It must be clearly understood that the regulation of the speed within the revolution, by the fly wheel, is a different matter from the regulation of the average speed by the governor, through a continuous accommodation of the power of the machine to its load.

Example 39. — Find the weight of fly wheel at an effective radius of 36 in. which will regulate the 14 × 15—250 engine of Fig. 177 within 1 per cent.

Circumference of wheel = $6 \times 3.1416 = 18.85$ ft.

Velocity of rim = $18.85 \times \frac{250}{60} = 78.54$ ft. per sec.

$$\frac{V^2}{g} = \frac{6169}{32.16} = 191.8; \qquad f = 0.01.$$

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From Table 16, the greatest value of E is 3410 ft. lb.; then for W we get by Eq. (185)

$$W = \frac{3410}{191.8 \times .01} = 1725 \text{ lb.}$$

If this all goes into the rim, and we make the latter 12 in. wide, with cast iron at 450 lb. per cu. ft. the rim would be about 2.5 in. thick.

EXAMPLE 40. — An engine 24 in. by 48 in. at 80 r.p.m., has a wheel 18 ft. in diameter and weighing 12,500 lb. How close will be the regulation if the m.e.p. is 40.2 lb. and the phase ratio k is 0.16?

The work per revolution is

$$W_R = 2 \times 4 \times 452.4 \times 40.2 = 145{,}490 \text{ ft. lb.}$$

The work to be absorbed by the wheel is then

$$E = 145{,}490 \times 0.16 = 23{,}280 \text{ ft. lb.}$$

The velocity of the rim is

$$V = 56.55 \times \frac{80}{60} = 75.40 \text{ ft. per sec.}$$

Then

$$\frac{V^2}{g} = \frac{5685}{32.16} = 176.8$$

and

$$f = \frac{23{,}280}{12{,}500 \times 176.8} = 0.01056.$$

(*b*) EFFECTIVE RADIUS OF WHEEL. — A mass particle m, at the end of a radius r which makes n turns per second, has the kinetic energy

$$e = \frac{mv^2}{2} = 2\pi^2 mr^2 n^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (186)$$

The total kinetic energy of the fly wheel, the summation of the e's of all the particles, will then be equal to the product of a constant by $\int mr^2$: which integral is the polar moment of inertia of the mass of the wheel about its rotation axis. As always in such a case, an equivalent effect would be got by imagining the whole mass of the wheel to be concentrated in a ring at the end of the radius of gyration: then this radius of gyration is the effective radius of the wheel, for which the velocity V in Eq. (185) is to be calculated.

Usually, the rim of the wheel contains by far the greater part of its mass, and the parts near the shaft, at short radius, have very little value as energy vehicles; so that no great error is caused, especially with wheels of the belt-pulley shape, by taking only the rim of the wheel into account, and using that as if concentrated at its outer circumference.

When, however, great accuracy is required, the polar moments of rim, arms, hub, and crank discs can be approximately computed and an equivalent mass at the outer radius found. And when there are

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other rotating bodies attached to the shaft, as for instance the armature of a generator, these also must be reduced to the wheel rim. The relation through which this reduction is made is that the energy value of any mass varies as the square of its radius from the axis, or that the mass for any energy value is inversely as the square of the radius.

(*c*) MULTIPLE-CRANK ARRANGEMENTS. — Besides using a fly wheel to restrain the fluctuations in speed due to irregularity in turning force, there is another method of securing uniformity of running: this consists in the use of two or more cranks at angles with each other (that is, not opposite, or at 180 deg). Then the excess phase of one crank can be made to coincide with the deficiency phase of the other; and not only will there be no dead center, so that the engine will start from any position, but there will be a much smaller variation in the total turning force or turning moment on the shaft. The freedom from dead centers is especially important in engines that have to start frequently against their full resistance, as locomotives, hoisting engines, and the like: and these are always made duplex, with cranks at right angles. Of course, a compound engine, with each cylinder driving its crank, will have the same action: except that in engines of the locomotive class, provision must be made for admitting steam directly to the low-pressure cylinder at starting, without waiting for it to get through the high-pressure cylinder.

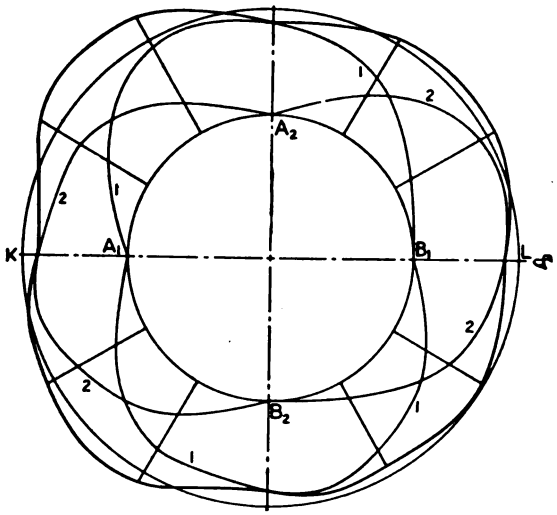

FIG. 178. — Circular Diagrams Combined.

An example showing the simplest case of this combined action is given in Figs. 178 and 179, which represent the working of a duplex

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simple engine with cranks at right angles. The two turning-force curves are supposed to be alike, and are taken from Fig. 175. The second crank will be at dead center when the leading crank is at 90 deg., hence the location of A_2 and B_2 on the drawings. The resultant curve of total turning force is got by adding the ordinates of the simple curves.

The circle or line of total resistance is drawn on the resultant curve, and it at once appears that the variations in driving force on the crank pin are much smaller, especially in comparison with the mean force, than in a single engine. The form of curve here shown, with a minimum at each quarter-point and a maximum near the middle of the quadrant, is characteristic of this type of engine.

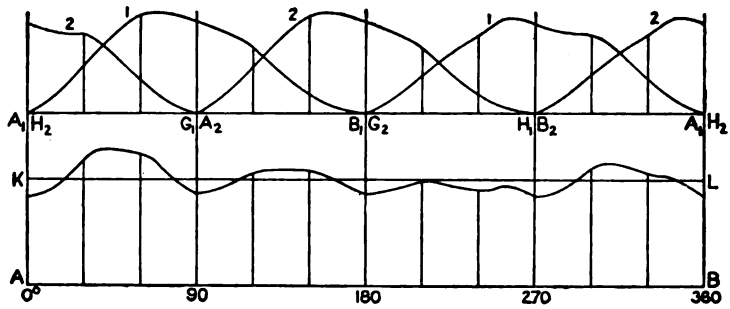

FIG. 179. — Turning Force in Quarter-crank Engine.

In the larger multiple-expansion engines, with three or four cranks, especially in marine engines, there is a considerable variety in the arrangement of the cranks, as to angles and as to order of sequence. Combined turning-force diagrams for these engines are made in the same manner as Fig. 178 or 179. An additional complication is encountered, in that the pistons are not of the same size; so that if the separate curves are to be combined, they must either be worked out for total force on crank, or else one piston must be taken as the chief and the pressures on the others be referred to that one, or be expressed as equivalent pressures per unit of area of that piston.

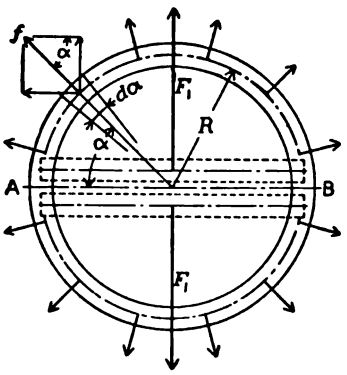

FIG. 180. — Centrifugal Force on Wheel-rim.

(*d*) STRESS IN RIM OF WHEEL. — The simplest possible problem as to the load imposed by centrifugal force is represented in Fig. 180, where the rim is considered a plain, homogeneous ring, the radial ten-

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sion of the arms being left out of account. The force tending to cause rupture at any section as AB, and resisted by tensile stress in the ring, is found by getting the value of the resultants F_1, F_1, each of which is the sum of the components, at right angles to AB, of all the radial forces on the particles of the half ring. The mathematical deduction of the value of this resultant force is as follows:

Let a be the area of cross section of the rim in square feet, l any length in feet measured along its circular center line, and w the weight per cubic foot of the material. Then the centrifugal force of a piece of unit length will be

$$f = \frac{aw}{g}\frac{V^2}{R}: \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (187)$$

and for an element of the length $dl = Rd\alpha$, we have

$$dF = fdl = fRd\alpha.$$

Now the component perpendicular to AB is $df \sin \alpha$, so that

$$dF_1 = fR \sin \alpha \, d\alpha:$$

and integrating for the half circle we get

$$F_1 = fR \int_0^{\pi} \sin \alpha \, d\alpha = 2fR. \quad . \quad . \quad . \quad . \quad . \quad (188)$$

The total centrifugal force on the half rim is

$$\tfrac{1}{2} F = \pi fR;$$

and we see that the bursting tendency is as if the force f were distributed along a bar of length equal to the diameter of the wheel, and that the force F_1 bears to the whole centrifugal force F of the rim the ratio $1 : \pi$.

This load is taken up at two sections of the rim, so that the tension at either section is $F/2\pi$.

EXAMPLE 41. — A wheel 16 ft. in diameter, with rim 24 in. wide by 3 in. thick, is made in sections and bolted together with five 1½ in. bolts at each joint: what will be the tensile stress in the cast-iron rim and in the bolts, on account of centrifugal force at 100 r.p.m., disregarding any holding effect which the arms may exert?

Using the mean diameter 15.75 ft., the velocity of the rim at 1.667 rev. per sec. is,

$$V = 3.142 \times 15.75 \times 1.667 = 82.47 \text{ ft. per sec.}$$

By computing the value of

$$\frac{V^2}{gR} = \frac{6801}{32.16 \times 7.875} = 26.86,$$

we see that the centrifugal force on each pound of rim mass is 26.86 lb.

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The weight of the rim is,

$$W = \frac{3 \times 24 \times 3.142 \times 189 \times 450}{1728} = 11{,}150 \text{ lb.}$$

And the tension on the rim is then

$$T = \frac{11{,}150 \times 26.86}{6.283} = 47{,}600 \text{ lb.}$$

On the 72 sq. in. of rim section this brings a stress of only 681 lb. per sq. in. On the five bolts, of an effective diameter (under the threads) of 1.25 in. and with 1.23 sq. in. of cross section, the stress will be

$$S = \frac{47{,}600}{1.23 \times 5} = 7750 \text{ lb. per sq. in.}$$

(*e*) LIMIT OF SPEED. — Manifestly, the analysis just exemplified is far from complete. In every wheel, the arms exert more or less radial tension, and the rim has some tendency to bend or bulge outward between them; the resulting bending stresses are accentuated when there are joints between the arms, especially with a thin rim. An attempt at a complete determination of stress, necessarily complicated and at the best only approximate, belongs to machine design rather than to the present course. However, in terms of the very simple assumption of Fig. 180, an important general relation between linear velocity and stress in the rim may be established.

Multiplying both sides of Eq. (187) by the circular length πD of the ring, we get

$$F = \pi D f = 2\pi R \frac{aw}{g} \frac{V^2}{R} \quad \ldots\ldots \quad (189)$$

Taking out the factor 2π, we have the tension T and can equate it to the expression for the strength of the ring, getting

$$T = \frac{aw}{g} V^2 = 144\, aS: \quad \ldots\ldots \quad (190)$$

from which

$$V^2 = \frac{144\, g}{w} S. \quad \ldots\ldots \quad (191)$$

It will be noted that in Eq. (189) the volume factor $2\pi R$ was necessarily in feet, while S is in pounds per square inch; and if we take a in square feet and w in pounds per cubic foot, the factor 144 must be introduced, as above.

For cast iron, with a maximum allowable working stress of, say, 4000 lb. per sq. in., the greatest safe speed would be

$$V = \sqrt{\frac{144 \times 32.16 \times 4000}{450}} = 203 \text{ ft. per sec.}$$

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While for a high-grade steel, where S might be as much as 20,000 lb., the value of V would be about $200. \times \sqrt{5} = 450$. This is greatly exceeded in turbines of the De Laval type; but there the radial tension of the solid wheel becomes a principal element of strength.

The reason for the disappearance of R from Eq. (189), expressed in other than purely mathematical terms, is that while, for a given linear velocity V, the centripetal acceleration varies inversely with the radius R, on the other hand the weight of a ring of given cross section increases as R.

Because of uncertainties as to homogeneity of construction, together with unavoidable irregularities in the distribution of stress, it is usual to consider 100 ft. per sec. as about the safe limit of speed for the rims of ordinary engine wheels.

§ 34. Pressures on Pins and Bearings

(a) Approximate Pressures on the Pins. — In Fig. 181, as heretofore, it is assumed that the mass of the connecting rod is divided into

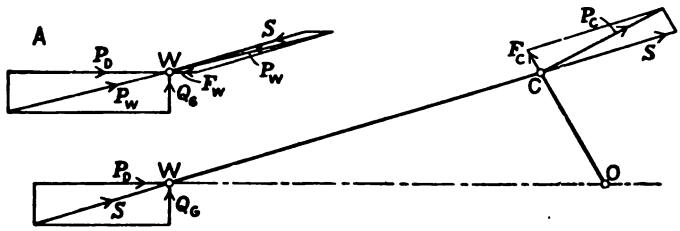

Fig. 181. — A Simple Determination of Pin Pressures.

two parts and concentrated at or around the pins. The portion at the wrist pin may be added to the slide mass and given its effect in determining the resultant driving force P_D. The rod, being now an ideal bar without mass, can have force acting only along its center line; wherefore the resultant of P_D and of the guide reaction Q_G must lie along the line WC. Up to this point, the diagram is identical with Figs. 173 and 174; but when the rod force S is carried over to the crank pin, instead of finding its moment or resolving it, we combine it with the centrifugal force F_C of the rod mass around the crank pin. The resultant is the pressure P_C on that pin.

A closer analysis of conditions at the wrist pin is made at A, Fig. 181. The driving force P_D is determined by the inertia of the purely sliding parts, the rod inertia F_W being kept separate. Center-line force S and inertia F_W combine to produce the pressure P_W of rod on wrist

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pin; and the resultant of P_D and Q_G must be the equilibrant of that pressure.

In Fig. 172, the value of F_0/A is 36 lb., due to a reciprocating mass which includes half the rod. Of the total weight of reciprocating parts, a good average assumption gives the piston slide 0.6 and the connecting rod 0.4: then one-half of the rod equals one-third of the slide, and out of a total of 36 lb. the half rod will have a centrifugal force of 9 lb. per sq. in. of piston.

(*b*) PRESSURE DIAGRAMS. — Fig. 182 shows successive pressures on the crank pin, got by the method of Fig. 181. At any particular posi-

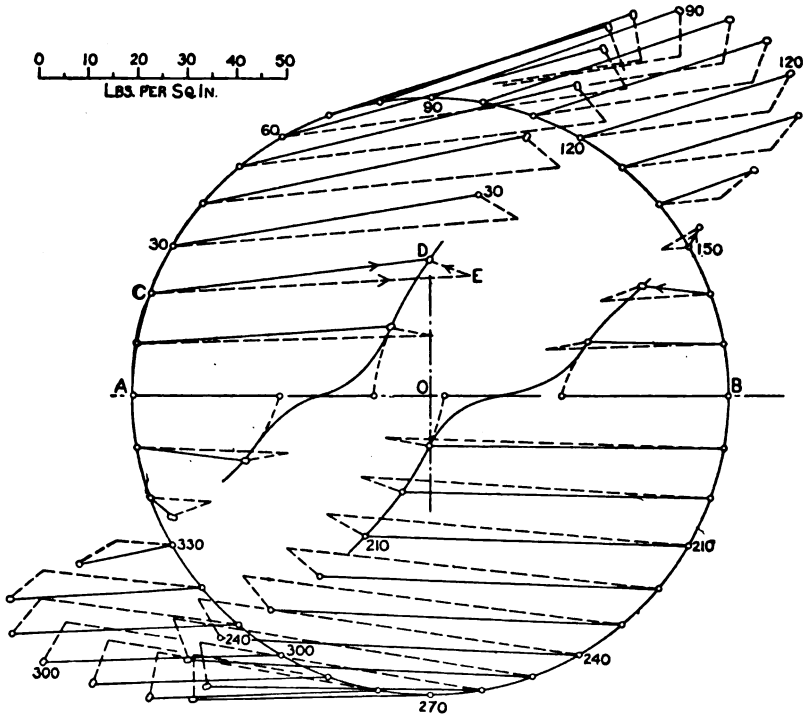

FIG. 182. — Diagram of Crank-pin Pressures.

tion, the line CE is the force S along the rod (found from the values of P_D in Fig. 172) and ED is the centrifugal force F_C, so that CD is the pressure P_C of the connecting rod upon the crank pin. It will be noted that some of the D points are double: in reality the full CD lines in this figure were found by an exact method, with a true determination of the inertia of the rod; and the fact that the discrepancies between the D points at the ends of the CD lines and those at the ends of the

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ED lines are scarcely visible is an ample proof of the sufficiency of the approximate method. Note further how the rod inertia F_C combines with the slant of the rod, in such fashion as to throw the pin pressures farther from the direction of the stroke line throughout the forward stroke, but to bring them almost into parallelism with it during the return stroke.

The simplest form of pin-pressure diagram, correct enough for all purposes of machine design, is given in Fig. 183. The effective steam-

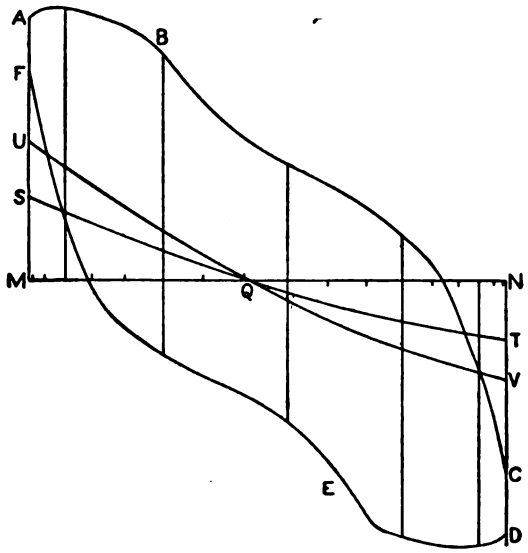

FIG. 183. — Approximate Pin Pressures.

pressure curves ABC and DEF are the same as in Fig. 172. By drawing the inertia-force curve ST for the slide alone, or for F_0/A = 27 lb. in this particular case, we get P_D as at A in Fig. 181, and this is practically the same in length as the wrist-pin pressure P_W. Again, drawing curve UV for the inertia of the slide plus the whole rod (for F_0/A = 45 lb.), we get the horizontal component of the crank-pin pressure P_C. Of course, ordinates are to be measured from ST or UV to the steam curves in either case. Inspection of Fig 182 shows that the difference between the crank-pin pressure and its horizontal component becomes relatively quite large when the pressures are small: but the values of the larger pressures are very closely given by the ordinates of Fig. 183.

(c) REVERSAL OF DRIVING FORCE. — Fig. 184 is intended to illustrate the effect of compression, and also of inertia, in causing less abrupt reversal of force in the engine and thus producing smoother

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running. The same forces are shown as in Fig. 183, but the base is changed to the developed crank circle, or the diagram is on a time base. In the plot of the effective steam-pressure curve, the vertical lines at CD and FA are reproduced from the stroke-line diagram (with apparently instantaneous change of pressure); but the actual manner of pressure rise in the filling of the clearance space will be as indicated by the short dotted curves. If there was no compression, the line of steam-force variation, toward and past the dead center, would be of

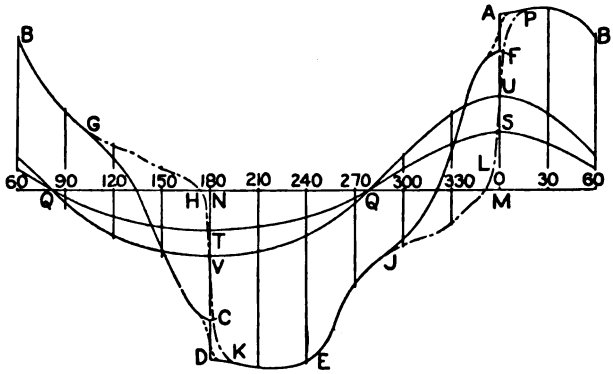

FIG. 184. — Developed Diagram of Horizontal Pin Pressures.

the form sketched at GHK and JLP. The speed of the pressure rise (from H to K and from L to P), and its exact timing in the cycle of the engine, would depend very much upon valve action in relation to volume of clearance; but it is quite sure to be much more rapid than the rise of back pressure caused by compression.

It is often said, rather loosely, that the mechanical effect or advantage of compression is "to bring the reciprocating parts quietly to rest," or "to absorb the shock of the reversal of motion." Neither expression is correctly descriptive. There is nothing of the nature of a "shock" in the smoothly fluctuating inertia force of the reciprocating parts; and while passing the dead centers this force is varying less rapidly than in any other part of the revolution. The spreading of the pressure reversal over a longer time, making it less abrupt and thus decreasing the tendency to shock in the taking up of any slack or looseness which may exist in the joints, is the true gain from cushioning by compressed steam. Further, as appears from the more exact representation of the crank-pin pressures in Fig. 182, if the reversal takes place far enough before dead center for the centrifugal force of the rod end to have a considerable vertical component, the pin will roll in the

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bearing while changing from side to side, not jump across the clearance gap; this seems to be about the only advantage in having the reversal come early.

In engines with a uniform rotary load, the exact timing of the force reversal is comparatively unimportant; but where the load is on the piston rod, as in pumping and blowing engines, it is well to have the development of forward driving force coincide in time with the development of resistance upon the working piston. Variation in the "lead" of the valve, or in the exact time of admission, is the important element of control in this connection.

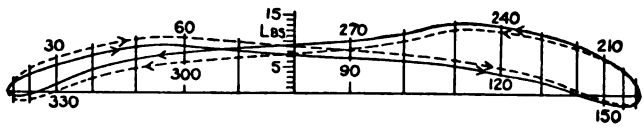

FIG. 185. — Diagram of Guide Reaction.

(*d*) GUIDE-BAR PRESSURES. — The curves in Fig. 185, on a stroke-line base, show how the guide reaction Q_G, in Figs. 153, 174, and 181, varies throughout both strokes: the ordinate from the base line to the curve represents the pressure of guide on crosshead. The full-line curve is gotten with exact rod inertia — see Fig. 154 for the general idea, but the method is not given in this book. The dotted curve shows Q_G as complementary to S, in Fig. 174; and both are for values of P_D from Fig. 172. The true inertia effect of the rod tends to throw the wrist-pin end outward from the stroke line, having the same general direction as the centrifugal tendency at the crank-pin end; the result is a lift during the forward stroke (decreasing Q_G) and a down-pull during the return stroke (increasing Q_G).

(*e*) PRESSURES ON THE SHAFT BEARINGS. — The method of § 30 (*e*) and Fig. 155 III gives an entirely satisfactory solution to the problem of finding the pressures on the main bearings of an engine: but as there presented it is applicable only to the simple case where all the force actions are symmetrical with respect to the plane of motion of the crank — that is, to a center-crank engine with equal wheels and with the same load forces on both wheels. The more general case of the ordinary side-crank engine is partly illustrated in Fig. 186.

Considering only the effective driving force P as acting upon the crank pin, this force is held in equilibrium by the two bearing pressures B_1 and B_2; the important dimensions being the overhang c of the crank pin beyond the middle of the main bearing O_1, and distance b between

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the latter and the outboard bearing O_2. Taking the origin of moments at O_2, we get B_1 by the relation

$$B_1 = \frac{b+c}{b} P; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (192)$$

and since B_1 is the middle one of a set of three parallel forces, B_2 is equal to $(B_1 - P)$: or, taking moments about O_1,

$$B_2 = \frac{c}{b} P. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (193)$$

Fig. 187 shows an extreme case of this extra pressure on the bearings; it is in correct proportions for a heavy locomotive, with long bearings

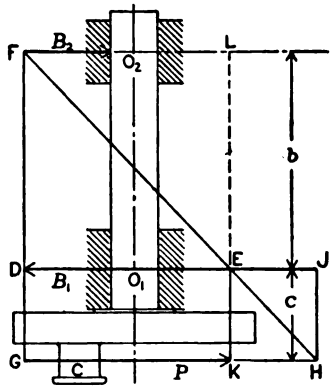

FIG. 186. — Bearing Pressures in a Side-crank Engine.

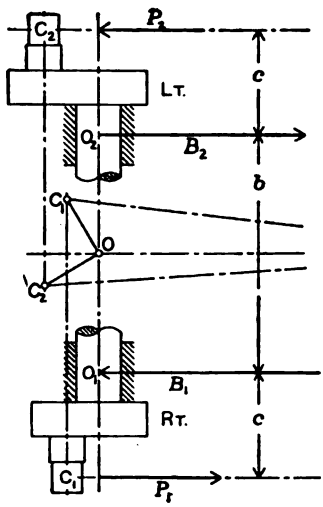

FIG. 187. — The Main Driving Axle of a Locomotive.

— necessarily kept within a definite limit of overall distance — and with the connecting rods outside the coupling rods. When, as here, the two forces P_1 and P_2 are opposite in direction, they act together in producing a turning moment which can be balanced only by much larger bearing pressures B_1 and B_2. In the portions of the revolution during which P_1 and P_2 point in the same direction, the B's are nearly equal to them.

(*f*) DIAGRAMS OF BEARING PRESSURE. — To illustrate the application of Fig. 155 III, as modified by the conclusions in Art. (*e*), we apply the force actions represented in Figs. 172, 182, etc., to the engine shaft outlined in Fig. 188, belonging to a side-crank engine with the armature generator mounted beside the wheel. The load being in the

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form of a torque only, if the armature is properly centered in the field, the forces to be combined are the crank-pin pressure P, the free counter-force F_B and the weight W. With the dimensions on the figure, these forces are divided between the bearings in the following proportions:

Force	at O_1	at O_2
P	$\frac{73}{59} = 1.24$	$\frac{14}{59} = 0.24\ (-)$
F_B	$\frac{68}{59} = 1.17$	$\frac{10}{59} = 0.17\ (-)$
W	$\frac{35}{59} = 0.59$	$\frac{24}{59} = 0.41$

These ratios are got by the method of Eq. (192).

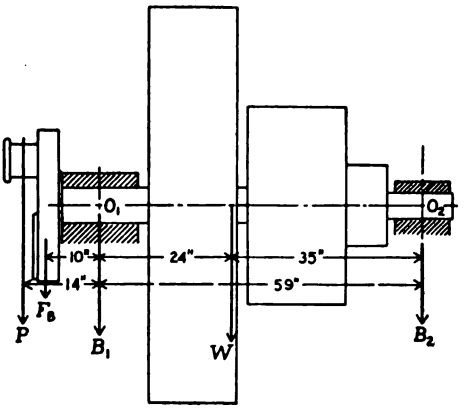

FIG. 188. — Outline of Engine Shaft.

In Fig. 189 is shown a series of force polygons like Fig. 155 III, taken at equal intervals around the crank circle, and determining the pressure B_1 at the main bearing of Fig.188. These diagrams, like all the others,

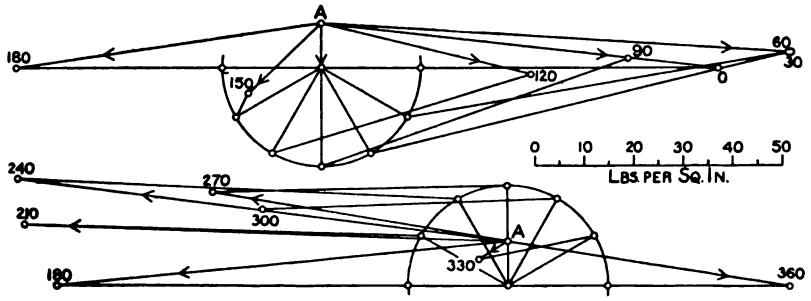

FIG. 189. — Pressures on the Main Bearing.

show force per square inch of piston. The weight, W/A, is about 15 lb., of which 9 lb. are carried at O_1, and 6 lb. at O_2; F_B/A is 17 lb., equivalent to $17 \times 1.17 = 20$ lb. at O_1; while P is taken from Fig. 182

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and multiplied by 1.24. The resultants give B_1, in the direction of action of shaft on bearing. It will be noted that the pressure on the crank pin is decidedly the predominating force, and that the prevailing direction of bearing pressure is nearly along the stroke line of the engine.

Similar diagrams can easily be drawn for the outboard bearing, keeping in mind the reversed directions of the components of P and F_B at O_2.

§ 35. Balancing the Engine

(*a*) FORCES AND MASSES INVOLVED. — As brought out in § 30 (*f*) and (*g*), the variable resultant of the inertia forces of all the moving parts of the engine is the force which tends to "shake" the body and foundation of the machine. Fig. 190 shows the masses and forces in-

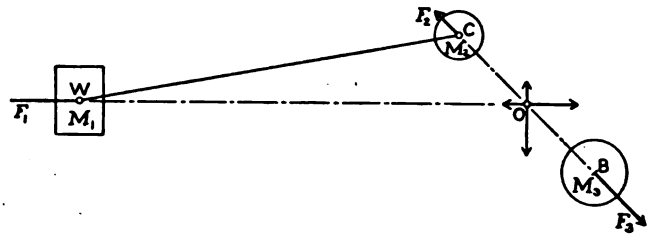

FIG. 190. — Accelerated Masses.

volved in the problem of balancing, or of reducing the resultant free force to the least attainable value. The masses are

M_1 = total reciprocating mass, including part of the rod.
M_2 = mass at crank pin, which may include, besides the rest of the rod, the crank pin and the crank arm or hub.
M_3 = mass of counterbalance, here taken at the radius OB, which is not necessarily the same as OC or R.

The proper way to divide the mass of the rod has been described in § 32 (*a*); the parts at the pins are made of such size as to keep the center of mass unchanged. With this division, there is exact equivalence to the real inertia force of the rod in amount and direction, but not quite the same line of action, since the resultant of the two partial forces must pass through the shaft axis. The proof of this statement is simple when the inertia action of the rod has been fully worked out, but in the absence of that analysis cannot be given here.

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The slide inertia F_1 is a component of F_0, the ideal centrifugal force of M_1 according to Eq. (172), or

$$F_1 = F_0\left(\cos\alpha + \frac{R}{L}\cos 2\alpha\right).$$

As to F_2 and F_3, their resultant $F_B = F_3 - F_2$ (see Fig. 157) is the free counterforce whose horizontal component partly balances F_1. This component is

$$H = F_B \cos\alpha.$$

By subtraction, we get the horizontal shaking force to be

$$S_H = (F_0 - F_B)\cos\alpha + \frac{R}{L}F_0\cos 2\alpha. \quad . \quad . \quad . \quad . \quad . \quad (194)$$

The vertical shaking force is, very simply,

$$S_V = -F_B \sin\alpha. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (195)$$

(*b*) Diagrams of Shaking Force. — A circular diagram which will give S_H for any crank angle is derived from Fig. 168 by changing

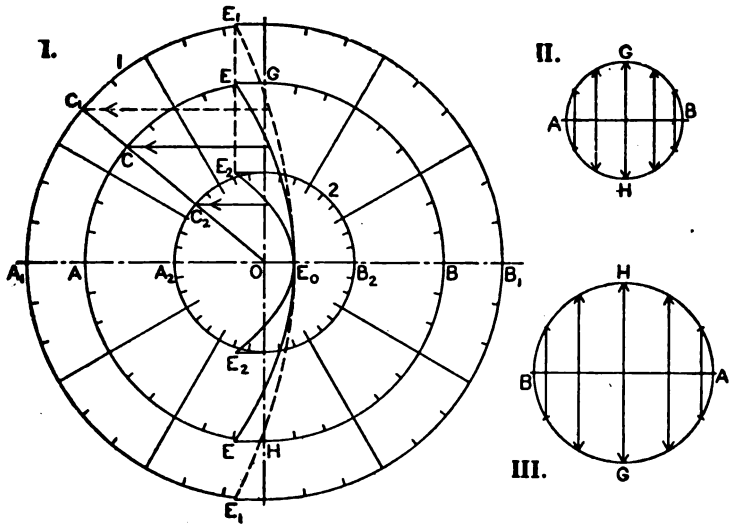

Fig. 191. — Shaking-force Components.

the radius of the circle to $(F_0 - F_B)$ without changing the distance of points on the curve E_1E_2 from the line GH. Thus, in Fig. 191 I, the diagram on AB is drawn with F_0 as radius, and gives F_1; which would be identical with S_H if M_3 were made equivalent to M_1, or F_3 equal to F_2 — that is, if the rotating masses on the crank were brought to a

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perfect balance. If F_3 were absent, F_B would be a negative quantity, and we should use the radius $OA_1 = F_0 + F_B$: with this circle No. 1, the E-curve is stretched out vertically to $E_1E_0E_1$. Diagram No. 2 is drawn for $F_B = \frac{1}{2} F_0$ and the E-curve is now squeezed together.

The amount of counterweight necessary to bring the crank disc, with its attached rotating mass (part of the connecting rod), to a state of centrifugal balance about O is well called the dead counterbalance: and the excess of F_3 over F_2 is then the free counterforce.

Similar diagrams for the vertical shaking force S_V are given at II and III. In the first, for no counterweight at all, the minus sign is neutralized by that of F_B, and the angle scale has its zero point A at the left, as usual: but where F_B has a positive value, we must measure α from the other dead center in order to get S_V, in direction as well as intensity, by direct measurement from the figure.

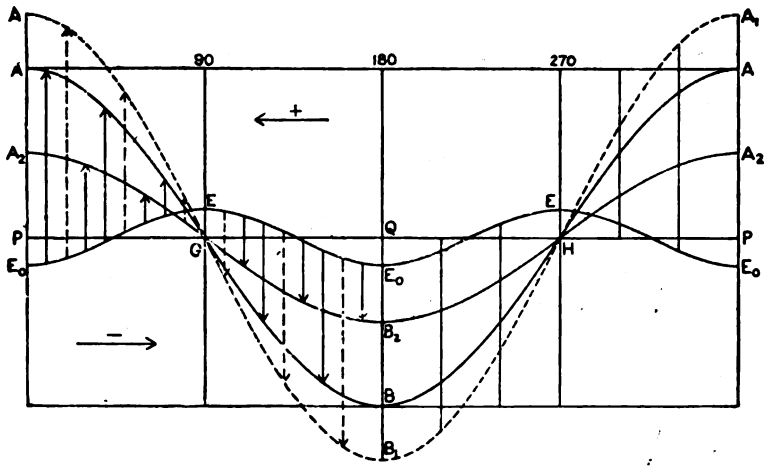

FIG. 192. — Shaking-force Curves.

Diagrams on the developed crank circle are shown in Fig. 192. Here the first term of Eq. (194), $(F_0 - F_B) \cos \alpha$, is laid off from the base PQP in one direction, in a simple sine curve for each value of $(F_0 - F_B)$; and the second term, $\frac{R}{L} F_0 \cos 2\alpha$, is measured from PQP in the opposite direction, in a sine curve of half the principal period. Then S_H is given by the ordinate measured from this E-curve to the particular AB-curve, as indicated by the arrowheads. It is made apparent that with a high degree of balancing — with F_B a large fraction of F_0 — the effect of the connecting rod, as shown by curve E_0E, becomes of greater relative importance.

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(c) THE SIDE-CRANK DUPLEX ENGINE. — The only engine truly represented by Fig. 190 is that with the center-crank arrangement, where the counterweights on the two cranks are symmetrical, so that they are equivalent to a single mass right in the axial plane. With the counterweight in another plane, relations exist analogous to those shown in Fig. 186. As covering this condition, and as a basis for the development of methods applicable to all multiple-unit engines, the mechanism of the common two-cylinder locomotive will now be considered. On the outline, in Fig. 193, only the free or active counter-

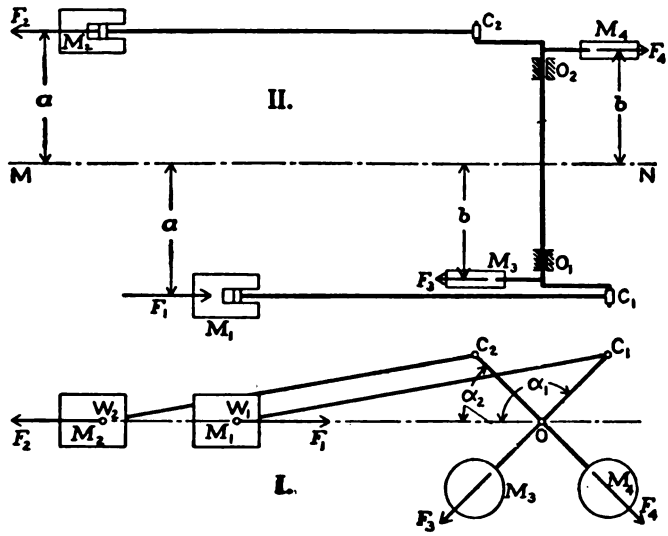

FIG. 193. — Inertia-force Outline of the Duplex Quarter-crank Engine.

balance is shown. The problem is to get a combined effect of the slide inertias F_1 and F_2, and then test the balancing value of F_3 and F_4, taking account of the offset of the latter from the engine stroke lines.

Viewing the plan of the engine at II, we see that none of these forces act through (or in the vertical plane of) the center of mass, which presumably lies in the plane projected in the line MN. Then according to the principle illustrated in Fig. 155, each of them, as F_1, will have two tendencies: first, to give the center of the whole engine mass a direct acceleration, as though the forces were along the line MN or in the vertical central plane, producing what we will call a shifting motion; second, to give an angular acceleration about a vertical axis through the center of mass, exerting the moment F_1a and producing an oscillating motion.

To get the combined shifting tendency of the two slide forces, we

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imagine them to be transferred to the central plane, taking their algebraic sum as laid out at I in Fig. 193. Considering OC_1, at the right-hand side of the machine, as the principal or leading crank, whose angle α_1 determines the position of the mechanism, and taking the inertia force to be positive toward the left, we have

$$F_1 = F_0\left(\cos\alpha_1 + \frac{R}{L}\cos 2\alpha_1\right), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (196)$$

$$F_2 = F_0\left[\cos(\alpha_1 - 90) + \frac{R}{L}\cos(2\alpha_1 - 180)\right]$$

$$= F_0\left(\sin\alpha_1 - \frac{R}{L}\cos 2\alpha_1\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (197)$$

Adding these and dropping the subscript from α_1, we get, for the total force,

$$S_H = F_0(\cos\alpha + \sin\alpha), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (198)$$

the two rod effects neutralizing each other.

(*d*) GRAPHICAL DETERMINATION OF SHIFTING FORCE. — Now just as the principal part of F_1 and of F_2 — respectively $F_0 \cos\alpha_1$ and $F_0 \cos\alpha_2$ — is the horizontal component of an ideal radial force, so also is the sum of these two, $F_0(\cos\alpha + \sin\alpha)$, the horizontal component of a single radial force, which force is the resultant of the two F_0's. This is proven in Fig. 194, where OC_1 and OC_2 are the respective F_0's, and OC is their resultant: then DOC_1 is α, and

$$OD_1 = F_0 \cos\alpha, \qquad OD_2 = F_0 \sin\alpha;$$

also

$$OD_1 = C_2F = D_2D:$$

wherefore

$$OD = F_0(\cos\alpha + \sin\alpha) = S_H;$$

and OD is the horizontal component of OC. This OC is $\sqrt{2} \times F_0$ or $1.414\,F_0$, and the value of S_H is therefore

$$S_H = 1.414\,F_0 \cos(\alpha - 45°). \quad . \quad (199)$$

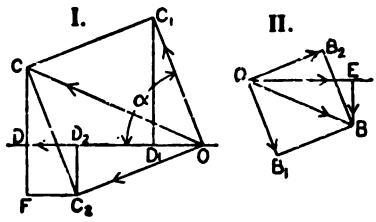

FIG. 194. — Combined Radial Effects.

The two counterforces, shifted to the center plane and represented by OB_1 and OB_2 in Fig. 194 II, can likewise be replaced by a single resultant OB, whose horizontal component OE will oppose OD, while the vertical component EB will be a free shaking force.

The results of this investigation are expressed in complete graphical form by Fig. 195. The circle in I is drawn with $1.414\,(F_0 - F_B)$ as radius, F_B standing for either F_3 or F_4 on Fig. 193. The zero of the

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angle scale is located at the position which OC, Fig. 194, will occupy when α for crank No. 1 is zero, or when OC_1 is on its zero dead center: and to get S_H we locate the actual value of α on this angle scale and measure over from the circle to GH.

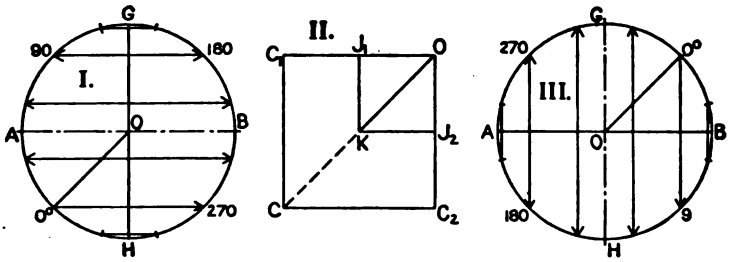

FIG. 195. — Diagrams of Shifting Force.

In Fig. 195 II, the diminution of OC_1 and OC_2 by C_1J_1 and C_2J_2, each equal to F_B, is shown with the purpose of giving a clear idea of the size of the primary forces. Then the resultant OK is the radius in I; while CK, the same as OB in Fig. 194 II, is used as radius in III. This second circle shows the vertical shaking force S_V in the same way that I gives S_H: its zero of angle is diametrically opposite to that in I, and the forces are measured vertically from AB.

(*e*) TORQUE EFFECT OF THE COMBINED INERTIAS. — Referring to Fig. 193 II, we note that F_1 and F_2 both act at the end of the lever arm a; and that when they point in the same absolute direction their moments oppose each other. The best way to combine them is to imagine F_2 to be swung, at the end of its radius a, through 180 deg. about some point on MN, until it comes into the line of F_1: then the algebraic sum of the two forces in this position — which is the same as their algebraic difference with F_2 in its actual direction — is a resultant free force acting at the end of the radius a to give the engine an angular acceleration about a vertical axis through the center of mass. Calling this turning or twisting force T_H, we have, without counterbalance, from Eqs. (196) and (197),

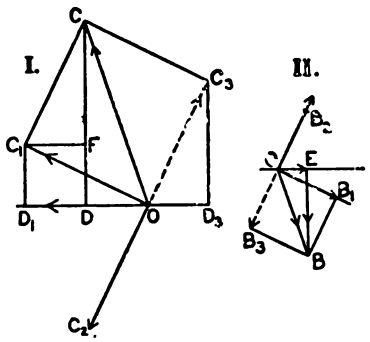

FIG. 196. — Radial Twisting Effects.

$$T_H = F_1 - F_2 = F_0(\cos\alpha - \sin\alpha) + 2\frac{R}{L}F_0\cos 2\alpha, \quad . \quad (200)$$

the two rod effects acting together in this combination.

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In Fig. 196, the reversal of F_2 in bringing it into the line of F_1 is represented by the reversal of the F_0 for the left side, from OC_2 to OC_3: then the major part of the force T_H is the horizontal component of OC: for

$$OD = OD_1 - DD_1 = OD_1 - D_3O = F_0 (\cos \alpha - \sin \alpha).$$

The two counterforces are combined in the same way at II, after reversing that for the left side, just as in Fig. 194 II: except that a reduced value of F_3 and F_4, or of F_B, must be used. That is, F_B at the radius b will be represented by a smaller force F_B' at the radius a, according to the equation

$$F_B b = F_B' a,$$

or

$$F_B' = \frac{b}{a} F_B. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (201)$$

This reduced force is laid off as OB_1 and OB_2 in II, b/a having the value 0.8.

To give the harmonic-motion component of the net inertia torque, a circular diagram like Fig. 195 I would have the radius 1.414 $(F_0 - F_B')$ and its zero at 45 deg. above OA. The rod-effect component might be got from a diagram similar to the curve EE_0E in Fig. 191, but a simpler method for that force is developed in the next article.

Considering possible oscillation about the three principal axes through the center of mass of a locomotive, the conditions as to plane of motion, axis of rotation, and turning force may be summarized as follows:

In horizontal plane, about vertical axis, net force T_H with moment arm a;

In vertical cross plane, about longitudinal axis, vertical component of OB, Fig. 196, with moment arm a;

In vertical longitudinal plane, about horizontal cross axis, net S_H (from Fig. 195 I) with distance from center of mass down to a plane through the two engine axes as moment arm.

In representing these moments each by a single force at the end of a lever arm from the center of mass, there must be the underlying idea of an equal and opposite inertia force (actual or equivalent), of the body mass at the center, which serves as the other force of a "couple."

(*f*) DETERMINATION OF THE ROD EFFECT. — The circular diagram as in Fig. 195 gives in very simple fashion the harmonic-motion or $F_0 \cos \alpha$ portion of the slide inertia. From the expression

$$f = \frac{R}{L} F_0 \cos 2\alpha = nF_0 \cos 2\alpha, \quad . \quad . \quad . \quad . \quad . \quad (202)$$

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for the component due to modification of slide motion by the connecting rod, an equally easy and general way of getting the resultant of several rod effects is derived in Fig. 197. At I, F_0 has the usual meaning, while f_0 is a similar ideal radial centrifugal force of the value $f_0 = nF_0$, so that Eq. (202) becomes

$$f = f_0 \cos 2\alpha. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (203)$$

The form of this equation suggests that f, being a sine function, can be shown by a plain circular diagram of the type that gives $F = F_0 \cos \alpha$

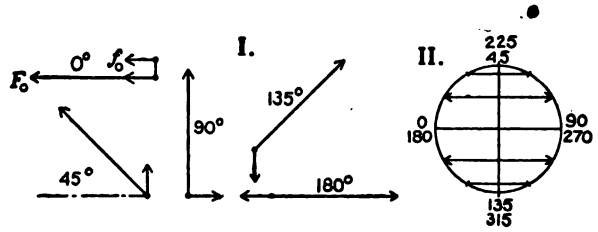

FIG. 197. — Circular Diagram of Rod Effect.

so readily; but it appears that the determining point on the circle, or the radius f_0, will have to rotate twice as fast as the crank. In illustration, five positions of the radial forces are shown, with f_0 making a complete revolution while F_0 turns through 180 deg. At 0 deg., f_0 is moved off to one side of F_0, with which it really coincides, for distinctness of representation. The circle in II is drawn with f_0, enlarged, as radius: and the use of the double angle scale, going twice around the circle, is self-evident. When the crank is at any angle α, the end of f_0 is at the correspondingly numbered point on the scale: and the distance to the vertical diameter is f.

(*g*) THE METHOD OF THE RADIAL RESULTANT is made complete by the expedient just devised, the application of which to the locomotive or quarter-crank engine is illustrated by Fig. 198. First, at I, the radial forces are laid out as if the two cranks were together at the zero dead center. When crank No. 2 is turned back through 90 deg. in order to get the actual arrangement, at II, f_{20} turns twice as far as F_{20}, so that the rod effects oppose and neutralize each other in shifting tendency. But when, for torque effect,

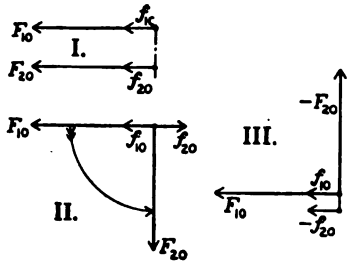

FIG. 198. — Rod Effect in the Quarter-crank Engine.

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we reverse F_{20} by the device of swinging it bodily into the plane of F_{10}, we likewise reverse f_{20}; and then, as shown at III, the rod effects combine. Of course, the radial resultant of the two F_0's will be used, as in Fig. 195.

Another example is given in Fig. 199: in rotating F_{20} through 180 deg., from the common position shown in Fig. 198 I, we turn f_{20} through 360 deg.; then the main components are self-balanced, and only the double rod effect is active for direct acceleration. Reversing F_{20}, at II, we see that the slide inertias act together to produce angular shake; so that unless the cylinders are so close together as to give a very short lever arm for the couple, counterbalancing may be necessary. The rod effects, however, neutralize each other in this action.

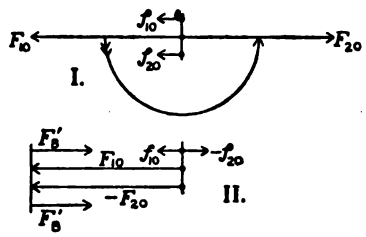

FIG. 199. — Radial Analysis of the Engine with Two Cranks at 180 deg.

In comparison with the trigonometrical method of Eqs. (198) and (200), this has the advantage of avoiding complicated formulas when the angles between the cranks are not quadrants; further, it obviates the need of interpreting the direction meaning of the algebraic sign of the angle functions, which is likely to be mentally confusing. It must be clearly understood that the radial resultants are imaginary, as are the F_0's and f_0's from which they are derived; only the components along the axis or stroke line of the engine are real forces. Lack of space forbids the further exposition of this scheme, but it is very effective for determining the resultant influence of the inertia forces of the moving parts in the more complex engines. Generally, such engines are largely self-balanced, so that there is little need to place counter weights on the cranks. In any case, the final result is best shown by developed diagrams similar to Fig. 192. With either type of two-crank engine, for instance, there will be three curves, one from each F_0 and F_B resultant (as from circles I and III in Fig. 195) and one from the resultant of the f_0's. As the reciprocating masses more nearly balance each other, the relative magnitude of the rod effect increases.

(*h*) EFFECTS OF THE SHAKING FORCE. — It is far easier to determine the unbalanced inertia force of the moving parts of an engine than to predict what effect this force will have in producing undesired motion of the machine as a whole: in fact, an answer to this latter question can be given only in general terms.

If the engine was supported in such a way that it could move freely in any direction, the recoil of the moving parts would give to it

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a motion similar to that of these internally-moving bodies, but smaller in the inverse ratio of the masses. By greatly increasing the "fixed" mass, or by bolting the engine to a heavy foundation, the amplitude of this motion is made very small. With earth-borne foundations, the shaking effect may be a mere tremor, hardly perceptible, a great deal depending upon the character and structure of the earth or rock upon which the foundation rests.

With a more elastic support, as in a ship or, occasionally, on the floor of a building, a large part of the structure will take up the motion of the engine bed, and the highest attainable degree of internal balancing becomes desirable. The worst disturbance is produced when the period of variation of the shaking force coincides with that of the elastic vibration of the structure. The latter is something that can be found only by trial.

In the preceding discussion of the combined effects in a complex engine, it is assumed that the engine is so compact and rigid that the several forces can be truly represented, as to their external effect, by a single resultant. Where the several parts are connected only by the foundations, as in many stationary engines of the "spread-out" type of construction, the separate force actions in the respective single engines are of greater interest than the combined effect. But engines in which the inertia forces are large — high-speed engines of any class — are usually of the close-constructed, self-contained type.

§ 36. Construction of the Engine.

(*a*) The Cylinder. — So far as the function of containing steam is concerned, the cylinder need be nothing more than a plain shell with heads bolted to the flanges, as it appears in the simple sectional view in Fig. 2. To provide for handling the steam, the valve chamber and steam passages must be added, greatly complicating the casting. A third requirement is the suitable support and holding of the cylinder, against its own weight and against the far greater working forces due to steam pressure.

Body of the Cylinder. — The main part of the inside surface, along which the piston slides, is called the bore of the cylinder; at the ends it is counterbored a little larger (from $\frac{1}{8}$ in. to $\frac{1}{2}$ in. on the diameter). This is done partly to facilitate reboring when worn, partly in order that the piston may not wear the rubbing surface to a shoulder at the end of its stroke: with the latter intent, the outer edge of the packing ring is even made to travel a little beyond the end of the bore in an engine like Fig. 200. The cylinder heads project into the counterbore,

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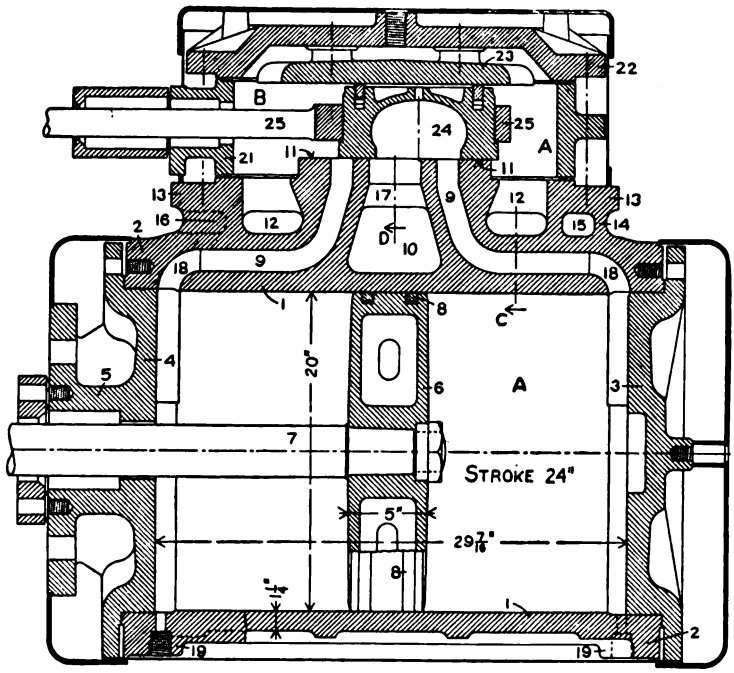

FIG. 200. — Lengthwise Section of Locomotive Cylinder, with short, wide, balanced slide valve and long ports. Scale 1 to 12.

1. Cylinder body or barrel.
2. Cylinder flanges.
3. Front cylinder head.
4. Back cylinder head.
5. Stuffing box.
6. Piston.
7. Piston rod.
8. Packing rings.
9. Steam ports.
10. Exhaust port.
11. Valve seat.
12. Steam-inlet ports.
13. Valve-chest base.
17, 18. Stiffening strutts across ports.
19. Drain-cock taps.
21. Valve-chest body.
22. Valve-chest cover.
23. Balance plate.
24. Valve.
25. Valve yoke and rod.

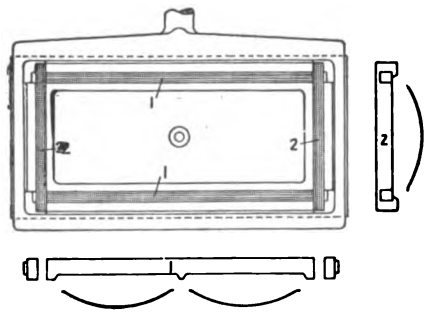

FIG. 201. — Top View of Valve and Yoke in Fig. 200. 1, 2, Balance strips.

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are conformed to the surface of the piston, and are stiffened by radial ribs on the outside. Fig. 200 shows them shallow, Fig. 203 very deep. We distinguish the full or plain outer head (at the head end) and the inner or stuffing-box head (at the crank end). In Fig. 203, with an extended piston rod, both heads contain stuffing boxes: here the heads are steam-jacketed, as well as the cylinder body, and the front head is cast with the cylinder. In the latter arrangement, it is the regular

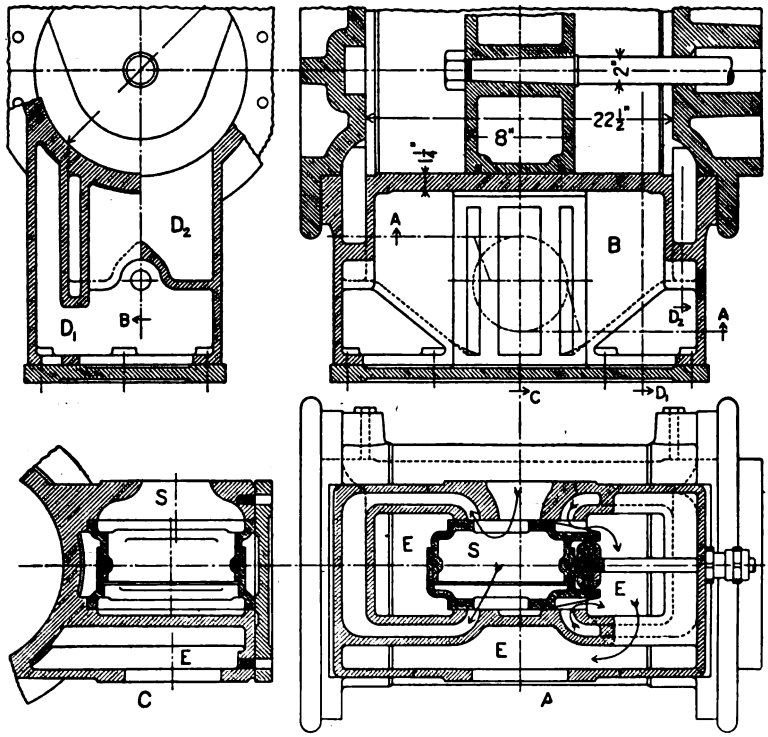

FIG. 202. — Cylinder of 15 in. by 14 in. High-speed Engine in Fig. 2, showing double-seated flat valve. Scale 1 to 14.

thing to make an opening large enough for the passage of a heavy boring bar, then close it by a bushing or small head which carries the stuffing box. Fig. 200 shows distinctly the narrow contact surface of the ground joint between head and cylinder; by concentrating the pressure due to the bolts on this narrow, accurately-finished ring surface, tightness is secured without the use of any packing. With plain flange faces, a gasket is commonly used, of heavy paper or very thin sheet packing, or of soft metal.

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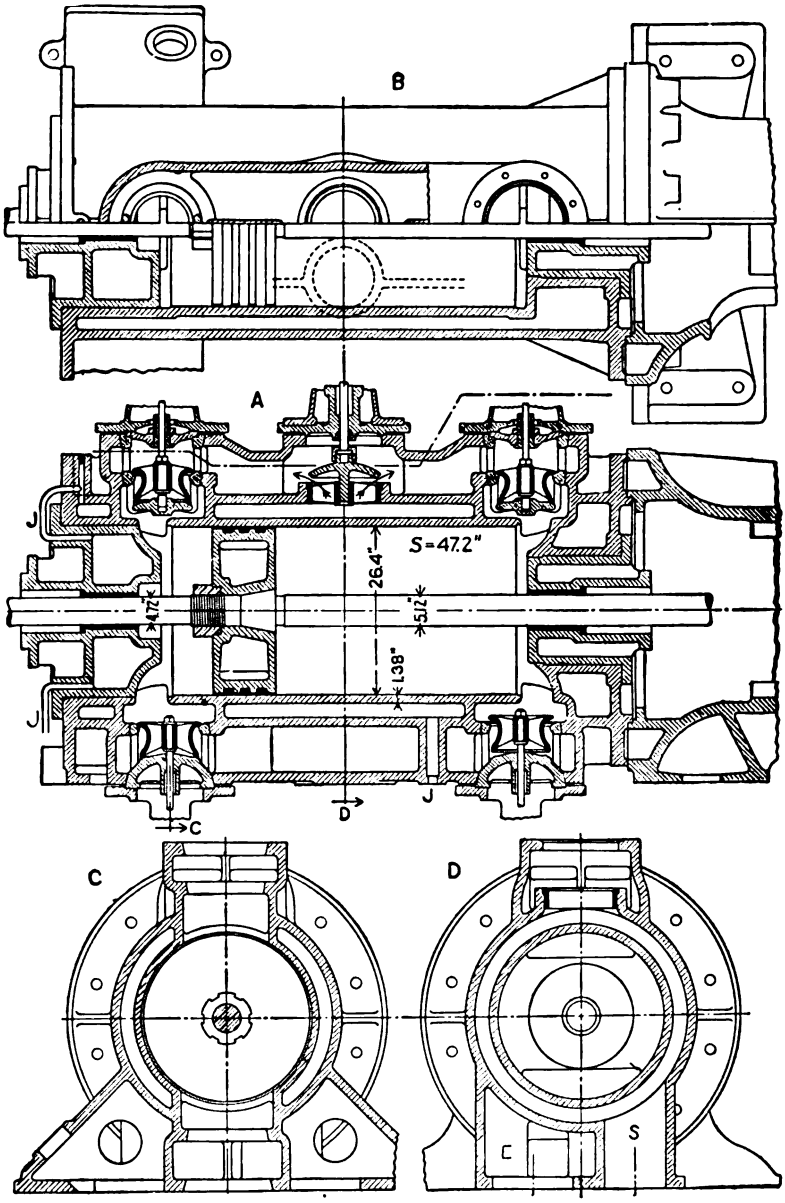

FIG. 203. — Lift-valve Engine, German Design; high-pressure cylinder of 670 and 1075 by 1200 mm. compound engine, with full steam jackets. Scale about 1 to 30.

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Formation of Steam Jacket. — In Fig. 203 is shown a jacket space cored in the main casting, a very common method of construction, especially in small and moderate sizes. To simplify the work of the foundry, both as to form of mold and in making it easier to get a sound casting, the steam jacket is often shut off by means of a separate inner shell or liner. This renders possible the use of a particularly hard and dense metal for the rubbing surface; but it adds joints which must be made and kept steam-tight.

Valve Chest and Steam Passages. — Fig. 200 shows an extreme type of the single slide-valve arrangement, the valve being short and the ports long; the valve chest is a separate box casting, and the joints between it and the cylinder and the cover are made tight by a copper-wire gasket; the valve detail in Fig. 201 will be referred to from the next chapter. In Figs. 9 and 259 we see the long slide valve, with short ports, and have representative examples of the two forms of balanced valve. Fig. 202 illustrates the cutting back of the ports into the cylinder-head space, as is markedly done in Fig. 203 also: the double-faced balanced valve is an excellent form, but requires rather long ports. Figs. 203 and 265 show four separate single-function valves, placed at the four "corners" of the cylinder.

The matter of valve form and action is taken up in the next chapter. Here it may be noted that the single slide valve, controlling two admissions and two exhausts (that is, the whole steam distribution at both ends) is properly called a four-function valve. Sometimes an engine will have one slide valve for admission and one for exhaust, each a two-function valve. The complete separation of function, as in Figs. 203 and 265, has many advantages in large engines, making the valve chambers smaller, simplifying the ports, and giving greater flexibility in the control of steam action.

Support of the Cylinder. — In the small high-speed, short-stroke engine, like Fig. 2, a single cylinder is bolted fast to the frame at the flange and allowed to overhang, its weight being relatively insignificant: if, however, the engine is a tandem compound, the outer cylinder is supported from the foundation. In vertical engines, this end support of the cylinder is all that is ever needed. Large horizontal engines have the cylinders resting on the foundation, as in Figs. 3, 203, and 265, a suitable base or footing being formed on the cylinder casting. The locomotive, represented by Fig. 200, is the only engine of current type that has its cylinder wholly supported from the side.

(*b*) FRAMEWORK OF THE ENGINE. — The bed or body of the small, compact, high-speed engine is well represented by the example in Figs. 2, 5, and 7; when a side-crank engine of this type has a separate outer

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bearing, or, in general, when the engine is direct-connected to a generator, there is usually an iron sub-base which extends out beneath this bearing, so that the engine does not depend upon the masonry foundation for alignment. The slower-running Corliss (or equivalent) engine has such a large wheel that the outboard bearing must be supported on a separate pier, which rises from the main foundation. The lighter and more open form of frame for such machine is outlined in Fig. 1, while Fig. 204 shows, to larger scale and in considerable detail, the

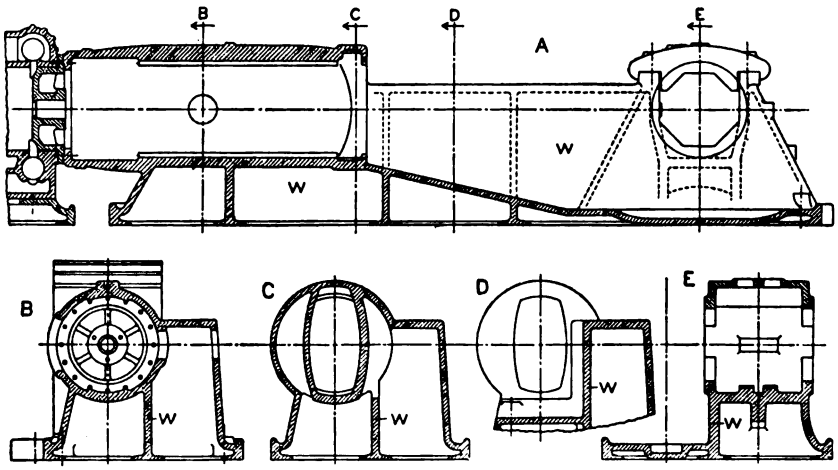

FIG. 204. — Frame of 26 in. by 48 in. Corliss Engine in Fig. 3. Scale 1 to 60.

frame casting of the modern, heavy-duty Corliss engine described in Chapter I; this is of the hollow, box form, with enclosed guides, and has a full-length bearing upon the foundation.

(*c*) THE PISTON. — Sectional views of the plain, hollow cast-iron box piston are given in Figs. 200, 202, 203, and 259: this is overwhelmingly prevalent for diameters up to 24 in., and is used up to as much as 48 in. Radial ribs are generally put in to stiffen the casting. In width of face or thickness of piston, Figs. 200 and 202 show extreme cases. Usually there are two plain packing rings, set near the edges of the face.

A number of single-disc pistons are given in Fig. 205. The cone disc at III represents the type mostly used in marine engines. The design at IV is special in having a cast-iron face ring on a cast-steel body: this gives a rather better surface as regards wear of the cylinder, and the ring is made broader at the bottom over 120 deg. of circumference, giving it more weight-carrying surface. An advantage of this arrangement is that when the cylinder is rebored only the face ring,

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not the whole piston, need be renewed. With all these pistons, the inner face of the cylinder head (or heads) is made to conform closely

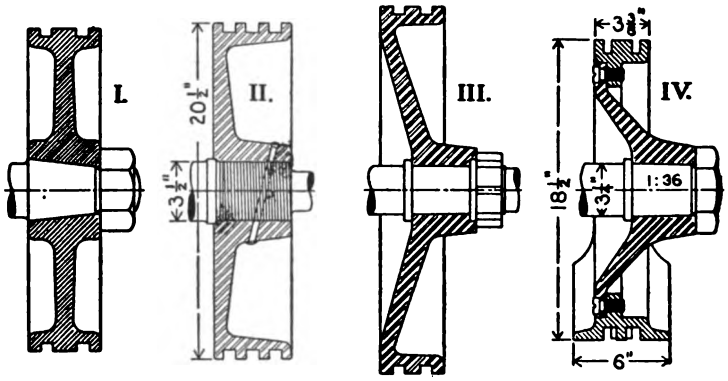

Fig. 205. — Locomotive Pistons of the Solid-disc Type. Scale 1 to 12.

to the shape of the piston surface, and thus keep down the clearance volume.

In Corliss engines, the built-up piston is usual, exemplified in Fig. 206. There is first the ribbed body or spider, fast on the rod. The

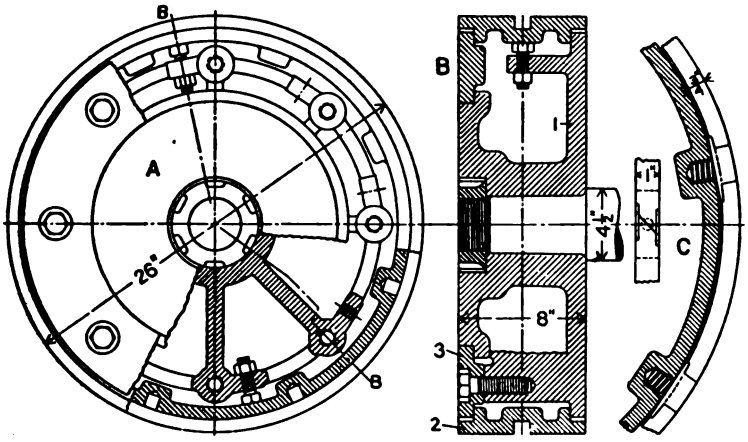

Fig. 206. — Piston for Corliss Engine in Fig. 3. Scale 1 to 12.
1. Body or "spider." 2. Bull ring. 3. Follower plate.

face or rim is formed by the bull ring, which can be adjusted on the body by set screws so as to get the piston rod into a truly axial position and keep it there as against wear of the sliding surfaces. Finally, the piston is closed by a follower plate, held in place by tap bolts.

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(*d*) PISTON PACKING. — The plain solid or one-piece, self-elastic packing ring, called a snap ring, is used in all small pistons and many large ones. Commonly of hard cast iron (sometimes of steel), this is turned to a diameter a little larger than the cylinder bore, then a piece is cut out of such length that when the piston is shoved into the cylinder the ends will just come together: a refinement of construction is to press the ends together, clamp the ring between two discs on a mandrel, and with a fine cut make it truly cylindrical when thus bent to the working diameter. Very often the ring is of uniform thickness; but it is better, especially with large diameters, to turn the inner surface slightly eccentric with the outer and make the cut on the thin side, thus getting a taper toward the joint. Very often the ends meet in a plain butt joint, but some form of lap is preferable because more effective to prevent leakage: in a horizontal engine the ring joints are usually kept at the bottom of the piston, but in a vertical cylinder the joints are placed opposite or far apart.

In Fig. 206 the one ring is in eight segments, which are pushed out by little coiled springs under the joint pieces. Such segmental rings are much used in large pistons, and are made in a great variety of forms. Occasionally auxiliary springs are used beneath one-piece rings.

(*e*) THE PISTON ROD. — The type of joint at piston and at crosshead is the chief point of interest in connection with this piece. The cone fit and nut, Figs. 2, 200, 203, etc., is most common at the piston: a straight fit is shown in Fig. 205 III, screw threads in the piston in Figs. 205 II and 259. With a very long taper, say from 1 in 20 to 1 in 40, it is important that the piston be brought up against a shoulder, to limit the wedge action of the rod; this shoulder may be formed by a reduction of diameter, as in Fig. 206, but it is better to have a collar on the rod, as in Fig. 200 or Fig. 205 IV.

At the crosshead, the screw and jam-nut joint seen in Figs. 2, 208, 209, and 210 is almost universal; except that in the locomotive the cotter shown by Fig. 211 is regularly used.

(*f*) PISTON-ROD PACKING. — For use in stuffing boxes, around piston and valve rods, soft or fibrous packing was long the only material employed; and it is only under more severe conditions of service and pressure that metallic packing has been coming into general use. Soft packings are made of vegetable fiber (hemp, etc.), asbestos, and rubber, in various combinations, with graphite frequently incorporated as a lubricant: there is on the market a host of these, ready-made in uniform and graded sizes, so as to fit and fill neatly the annular space around the rod.

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As to the stuffing box, we note that the end surfaces may be either flat as in Fig. 9 or conical as in Fig. 265. To allow for initial or acquired faults in alignment and for wear of the sliding surfaces of piston and crosshead, the rod must fit but loosely where it passes through the cylinder head and the gland. Quite often bushings of brass or linings of babbitt metal are used, as seen in Figs. 2 and 203.

A typical metallic packing is shown in Fig. 207. The gland G is here a mere heavy cover plate, made tight by a copper-wire ring basket. The ball ring 1 is seated upon the gland with a spherical

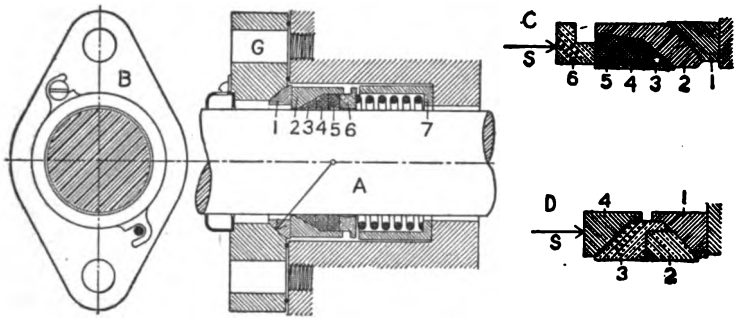

FIG. 207. — Metallic Piston-rod Packing, to fit stuffing box in Fig. 200.

ground joint; the casing 2 is free to move sidewise, and contains the three babbitt-metal packing rings 3, 4, 5, which are the only parts that touch the rod. These, as pieces requiring comparatively frequent renewal, are made in segments so that they can be put in without disturbing the rod connections, and are placed in the cup so as to "break joints." The follower 6 is pushed up by the spring, which is itself held in the light casing 7. This spring is not expected to do much more than hold the rings in place: it is the steam pressure that wedges the rings into the cup and makes the joint tight, so that the tightness of the joint varies with the pressure of the steam to be held.

The same packing, with different proportions, is shown in detail at C — the difference consisting in the slant of the confining surfaces of the cup, which causes all the rings to come into action, although the first one still takes most of the wear. And at D is given the detail of another packing of this same wedge-action type, with brass rubbing rings.

(*g*) THE CROSSHEAD. — The three typical forms of this piece are, four-bar or wing crosshead of Fig. 208, the slipper type in Fig. 209, and the block or trunk form in Figs. 210 and 211. The first requires four guide bars, two at each side, and is little used; the second has but

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one main surface, although top bars extend inward over the edges of the flat soleplate, to prevent lifting of the crosshead; the third form,

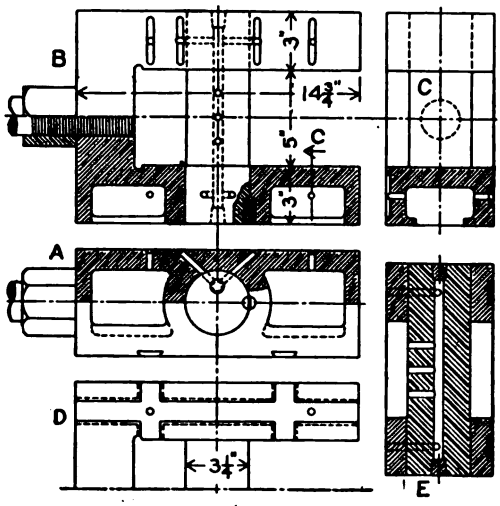

FIG. 208. — Crosshead from Engine in Fig. 2, of four-bar type. View D shows grooves in bottom face which are filled with babbitt metal, so that rubbing surface is partly babbitt, partly cast iron: view E shows how wrist pin is lubricated by oil caught from guides. Scale 1 to 10.

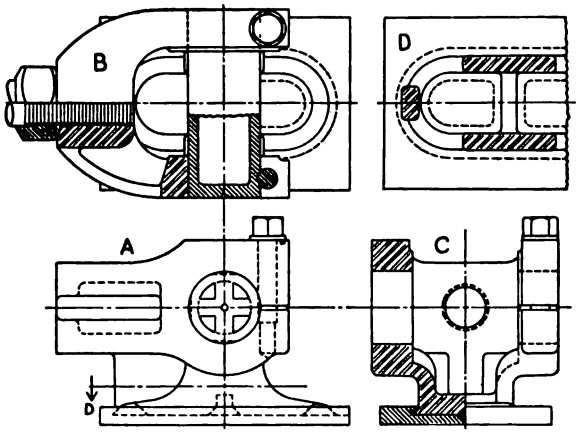

FIG. 209. — Crosshead of Slipper Type, with cast-steel body and cast-iron soleplate; hollow cast-iron wrist pin.

with guides below and above in the plane of the mechanism, is the most used.

In general, any crosshead has the two functions of joining the connecting rod to the piston rod and of sliding upon or between the guides:

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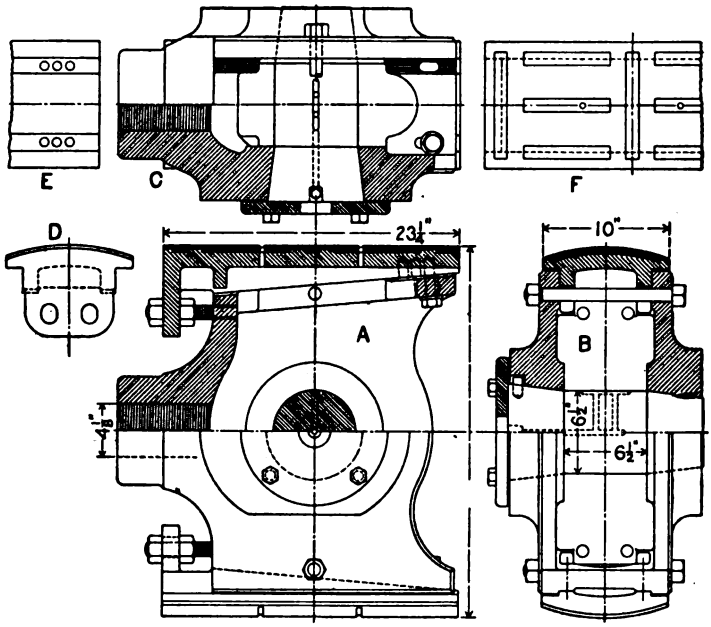

FIG. 210. — Crosshead of Block Type, for bored or cylindrical guides, as in Fig. 6, and with adjustable wedge shoes. View E shows inside of shoe, with holes for cap screws; view F shows dovetail grooves for holding babbitt facing.

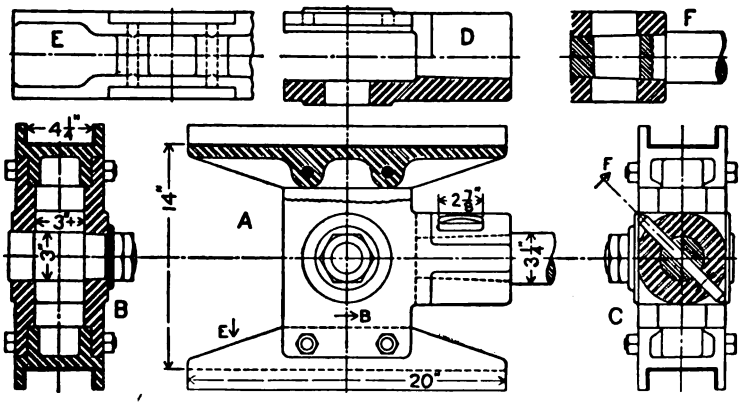

FIG. 211. — Block Crosshead of the "Alligator" Form, commonly used in locomotives; this to accompany cylinder in Fig. 200. Body and shoes of cast steel; rubbing surfaces faced with tin, which is self-soldered to steel. Scale 1 to 12.

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it consists therefore of the body, which begins with the hub for the piston rod and is usually forked to receive the wrist pin, together with the sliding faces or shoes, of whatever form. As to the wrist pin, cylindrical fits are shown in Figs. 208 and 209, taper fits in the other two examples. In Fig. 208 the pin is driven in tight and the rod has a strap end; usually, however, the wrist pin is made easily removable and this is essential with a solid-end rod. The figures show different kinds of rubbing surface, but only the one example in Fig. 210 (characteristic of large, stationary engines) has provision for adjustment to take up wear.

(*h*) THE CONNECTING ROD. — This piece may be analyzed into a shank or body and two ends or heads. In large, slow-running engines the shank is usually round, largest in the middle and tapering toward the heads, as in Fig. 212; at higher speeds this type changes to one long cone, tapering outward toward the crank end and flatted on the sides so

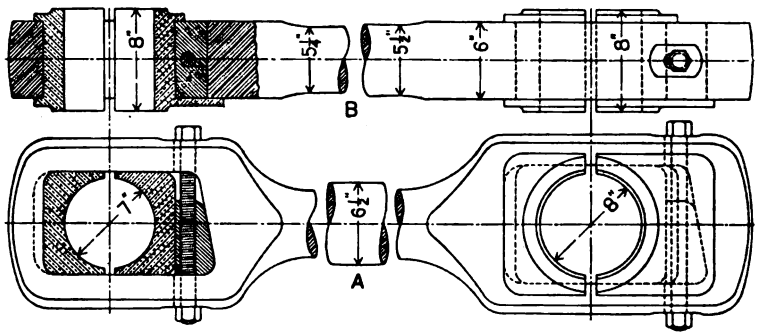

FIG. 212. — Connecting Rod with Solid Ends, belonging to Corliss engine in Fig. 3. Scale 1 to 15.

that it approaches a rectangular section as the diameter increases. Stationary high-speed engines (Fig. 2) and many locomotives have the rectangular section, increasing in depth toward the crank end. For fast locomotives the sides of the bar are milled out, giving the I-beam section in Fig. 214. These differences are dictated by the need of providing resistance to the transverse inertia forces which result from the swing of the rod. In stationary engines, the ratio of rod length to crank radius is commonly 6, or somewhere near that value; in locomotives it ranges from 6 to 10, and in marine engines is usually about 4.

At each end of the rod there is an adjustable bearing for one of the pins. The "boxes" or "brasses" which form the bearing proper must be enclosed in a frame or casing of suitable form and strength, and provision must be made for setting them to a proper fit upon the

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pin and for taking up wear. Four types of rod end may be here differentiated: the solid end is simplest in construction, but it cannot be used with an inside crank, and the boxes have a minimum of holding flange on one side, as appears in views C and D of Fig. 213. The bolted

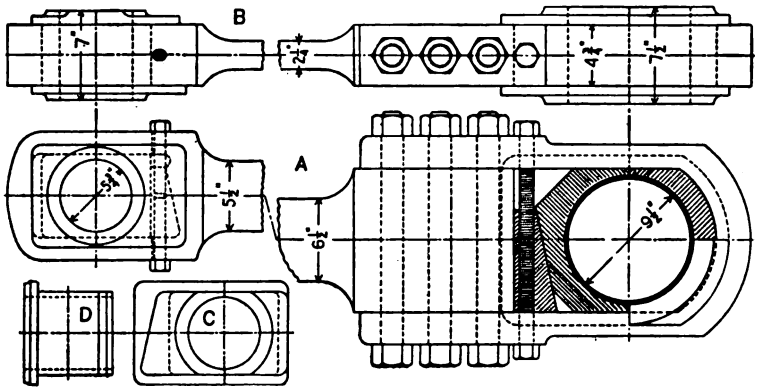

FIG. 213. — Rod with Bolted Strap at Crank-pin end, for 22 and 42 by 27 in. cross-compound engine at 175 r.p.m. Scale 1 to 15.

strap is a very good type of end construction, while the essentially equivalent jaw end in Fig. 214 is based on German practice. The marine

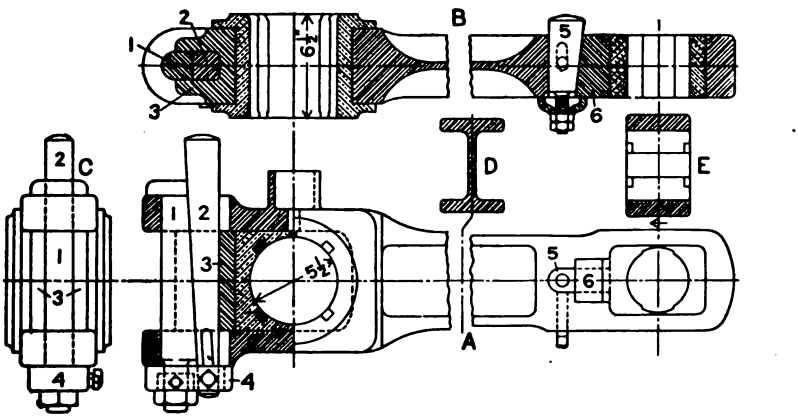

FIG. 214. — Rod for Locomotive, with special forked end. Scale 1 to 12.

type, at the crank pin in Fig. 2, is used almost without exception on marine engines, and very largely in stationary practice.

A point to be noted is whether adjustment for wear will shorten or lengthen the rod. In Figs. 2, 212, and 214 the two take-ups tend to balance each other, but in Fig. 213 the two work together to increase the length between centers. Except in the very severe locomotive

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service, crank-pin boxes are generally lined with white metal, being commonly made of cast iron or steel; at the smaller wrist pin, with much less motion in the joint, brass boxes are almost universal.

(*i*) THE CRANK SHAFT. — The usual type of shaft for stationary side-crank engines is well represented by Fig. 215 I. Shaft and crank pin are of mild steel, the disc is of cast iron with a fan-shaped counterweight formed upon it. They are put together with either forced or

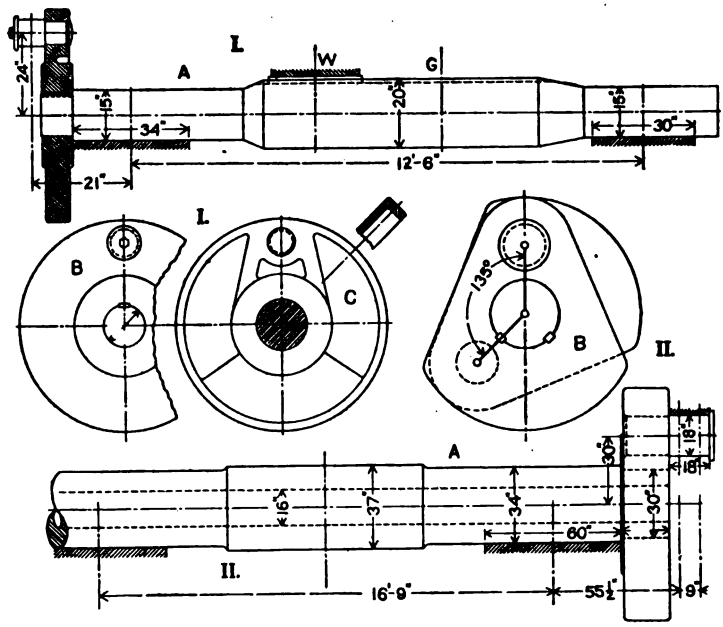

FIG. 215. — Large Shafts with Built-up End Cranks. I, for engine in Fig. 3, scale 1 to 40; II, for 44 and 88 by 60 in. duplex horizontal-vertical Corliss, like that in Fig. 137, rated 8000 i.h.p. at 75 r.p.m. Scale 1 to 72.

shrunk fits, the holes in the disc being bored about one in one thousand smaller than the pieces which are to go into them. The crank pin is riveted over for greater security, and has a detachable cap, to receive the solid-end connecting rod shown in Fig. 212. To carry the weight of the wheel and generator, the shaft is enlarged between the bearings; but its diameter can be reduced where it enters the crank hub without sacrifice of needed strength, although the proportion of reduction is greater in this case than where the conditions of working are more severe.

Figure 215 II shows the shaft for a large duplex horizontal-vertical engine. The straight shaft is a hollow forging of high-grade steel, oil-tempered; the discs or webs are massive steel castings. Referring to

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the outline in Fig. 137, we see that the two connecting rods act upon one pin, as here indicated by the letters H and V.

A typical shaft for a high-speed center-crank engine is shown in Figs. 2 and 7. To use a complete disc as in Fig. 215 I neutralizes a good deal of the off-center mass, and the fan-weight form is therefore more effective. Since the crank is enclosed, appearances do not suffer, and the connecting rod is more accessible than if shut in between circular discs. When a greater counterforce is needed than can conveniently be provided with iron, the "weights" are made hollow and filled with lead, especially on locomotive wheels.

Beside large built-up arrangements as in Fig. 215, solid-forged shafts are much used in marine engines, more on naval vessels than in merchant service.

(*j*) Bearings. — Fig. 216 shows a number of bearings from small high-speed engines. The arrangement at I is simple and very satisfactory as long as the bearing surface remains in good working order;

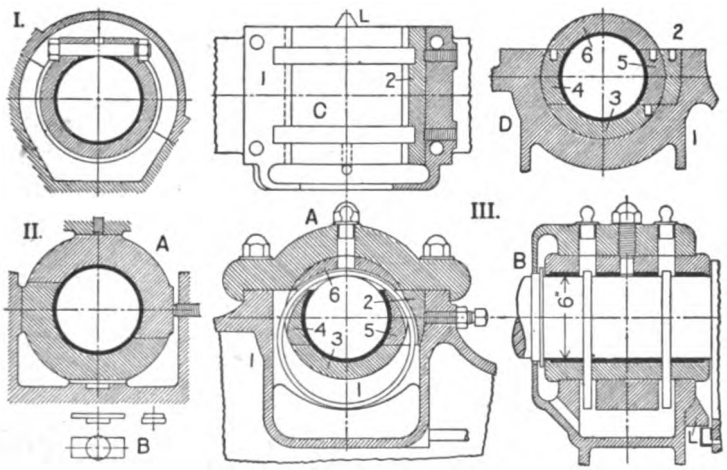

Fig. 216. — Bearings for High-speed Stationary Engines. I, one-piece bushing, with elastic adjustment; II, simple bearing in two parts; III, ring-oiling bearing in four parts.

but the fact that a wheel must be taken off in order to replace a damaged bearing shell has led to the substitution of a two-piece bushing, with a solidly-bolted joint at the bottom.

A very simple two-piece bearing is shown at II. The main set-screw adjustment is at the right, while that on top obviates the need of close fitting of the cap on the bearing boxes. With only a slight side-wise displacement of the shaft, the lower box can be easily taken out

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for examination or repair. Beneath the main view is a detail of the universal key which fits into a round hole in the bed and a cross-slot in the bottom box, so as to hold the latter against endwise movement.

At III in Fig. 216 a four-part bearing with oiling rings is drawn more in detail. The adjustable quarter box 5 is backed by a face block 2, of which the outer surface is cylindrical below the level of the shaft axis, so that it can be easily taken out. In view D are shown the holes for lifting screws which are tapped in the several parts of the bearings. In the middle of the bearing, at the top, is a screw pin which serves as a dowel, to keep the boxes from ever turning with the shaft. View C is a plan of the casing, with the boxes removed. The lubrication arrangements, besides the oil well and the rings with peep holes above them, include a light collar fastened upon the shaft at the outer end of the bearing, to catch all oil that escapes and return it to the well. At the inner end the oil drips upon the projection L, from which it is scraped by a little catcher on the crank disc, and carried to the crank pin.

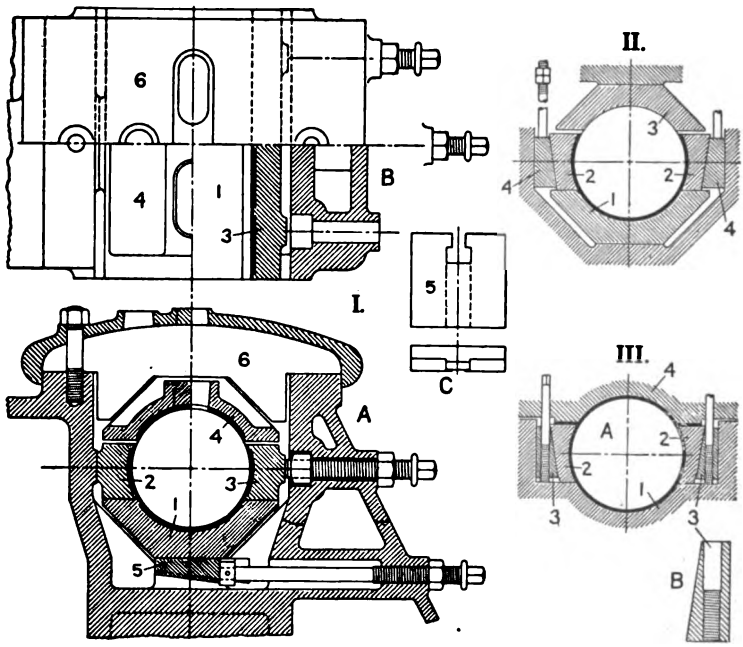

FIG. 217. — Bearings for Engines of the Corliss Type. I, from engine in Fig. 3, scale 1 to 24; II, III, different arrangements of adjusting wedges.

Corliss-engine Bearings. — In horizontal engines of the Corliss type, the bearings are usually made in four parts, with side adjustment either

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by set screws or by wedges. Fig. 217 I shows vertical adjustment also, by means of the wedge drawn in detail at C. It is hardly possible that a lifting force greater than the weight of the shaft and wheel will ever be developed in an engine of this class; consequently, the top box is made light, and is held down simply by contact with the cap at the ends. The big hollow cap is characteristic. In view B, the upper half is a plan of the bearing cap, and the lower half is partly a top view of the base, partly a section by a plane through the shaft axis.

Figure 217 II shows a double-wedge adjustment, the wedges being drawn up by long studs, with nuts on top of the cap. With this arrangement in only one of the two bearings, it is possible always to square the shaft with the stroke line, besides taking up wear — provided the bottom box can move, as in this case. Where the lower part of the bearing is solid with the frame, as at III, the wedges can be used

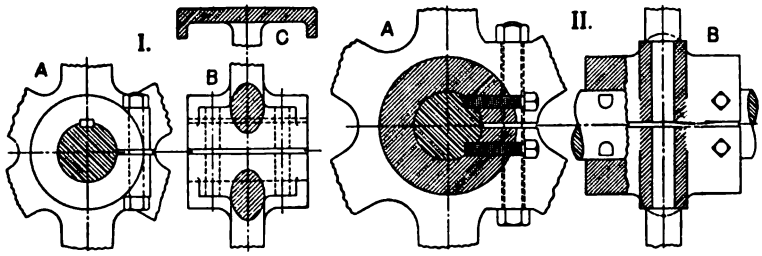

FIG. 218. — Fly-wheel Hubs for Small Engines. I, standard design; II, for engine in Fig. 2. Scale 1 to 16.

only to adjust the side boxes to a proper fit. These wedges are raised by set screws, which go through them and rest upon the seatings beneath them, so that the cap can be taken off without affecting the adjustment.

(*k*) WHEELS. — The two typical forms of engine wheel are very well shown in Figs. 2 and 7 and in Fig. 3. Stationary engines for general service are made with wheels of the belt-pulley type, and in the smaller sizes this is commonly retained when the engines are direct-connected to generators. For larger machines, direct-connected or directly loaded, the balance-wheel type with rectangular cross section is usual.

Small belt-pulley wheels are made with inside flanges on the rim, as in Fig. 7. With diameters less than 9 ft. they are usually cast in one piece; but very generally the hubs are split, on one side or all the way through, so that they can be clamped upon the shaft. Fig. 218 II is a typical arrangement, with two bolts at one side of the hub and a common rectangular key. At II there is a single big bolt in the plane

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of the arms, and the key is replaced by set screws which fit into pockets milled in the shaft.

Wheels up to 16 ft. in diameter are commonly made in halves — the size of the largest piece that can be shipped on an ordinary railroad car having a good deal of influence in this matter. A representative example of the balance-wheel type is detailed at I in Fig. 219.

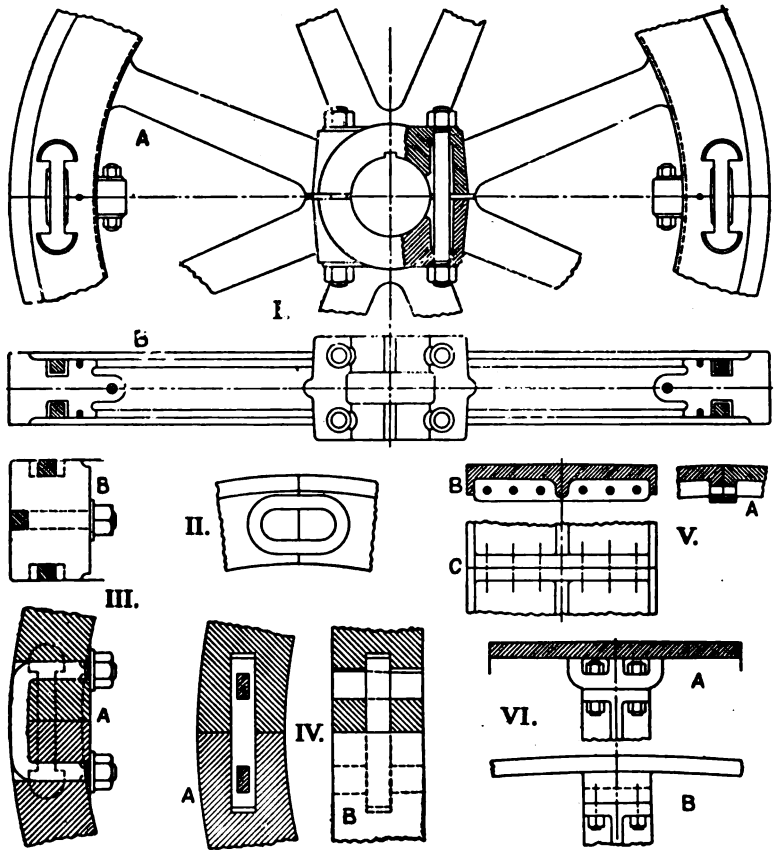

FIG. 219. — Balance Wheels of Medium Size. I, from engine in Fig. 3; diameter 16 ft., weight 62,000 lb., scale 1 to 48; II to VI, details of wheel joints.

The hub is strongly clamped upon the shaft by four heavy bolts, while the strongest part of the rim joint consists of the two I-shaped shrink bolts or links. Wheels of this size will be completely finished in the shop; and to insure a neat fit at the joints, little screw dowels are put into holes drilled and tapped half and half in the two parts. When the engine is erected, these will be inserted first, along with the light bolts

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through the lugs on the inside of the rim: and the shrink bolts are put in last of all.

Another form of connecting link is shown at II, the name "link" being here closely descriptive. At III a U-shaped tie is used along with I links. The tie bar with keys, Fig. 219 II, is sometimes used for low speeds and light stresses: this is not a shrunk joint, but is tightened by making the keys with a slight taper and driving them in hard.

Large wheels of the belt-pulley type are generally joined by bolted flanges, after the manner of the sketch at V. In large diameters, wheels of either type are made in a number of segments, each with one or two arms, and the arms are held in a built-up hub. Belt wheels have been constructed with the arms separate from the rim segments and bolted fast as in Fig. 219 VI (the arms having a cross-shaped section) but it is better to cast rim and arms together.

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CHAPTER VIII

VALVE GEARS AND GOVERNORS

§ 37. The Simple Slide Valve, with Harmonic Motion

(*a*) The General Form and manner of action of the slide valve and its driving gear are illustrated and explained in Chapter I. The first step in a closer study is to establish simple and convenient graphical methods for showing the movement of the valve, or for readily finding its position corresponding to any position of the crank or piston. The movement of the valve is determined by the rotating eccentric, just as that of the engine piston is determined by the crank,

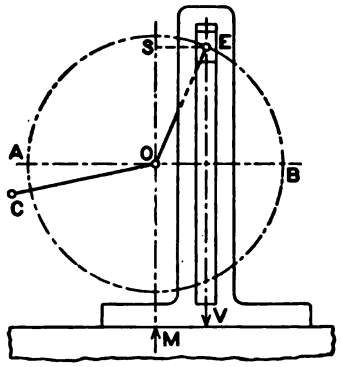

Fig. 220. — The Valve Mechanism Reduced.

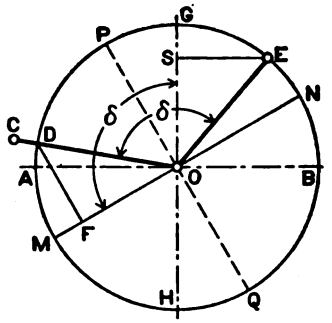

Fig. 221. — The Reuleaux Diagram.

and the deductions in § 31 apply equally to the valve gear. The present problem is simplified by the fact that the eccentric rod is usually so long, relative to the radius of the eccentric, that the valve receives practically harmonic motion: then, for kinematic study, the whole mechanism, including the main crank arm, may be reduced to the form shown in Fig. 220.

This is a modification of Fig. 158; and by placing the slide beneath the eccentric circle we emphasize the fact that the distance of the valve from its mid-position, or MV, is the same as the distance SE of the

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eccentric center E from the vertical center line SO. The position of the valve is defined by giving this distance, measured to the right or left: we shall call it the valve travel, and denote it by t.

(*b*) MOVEMENT DIAGRAMS. — In order to get the valve travel corresponding to any crank position, knowing the eccentric angle δ or COE in Fig. 220, we must measure forward this angle δ and find the length of ES or t. But a truly serviceable diagram should give t directly from the crank angle, without the bother of repeatedly laying off δ. One diagram meeting this requirement is derived in Fig. 221, where the figure made up of the reference line GH, the eccentric radius OE, and the t line ES is rotated backward about O through the angle δ. Then GH takes up the constant position MN, while OE coincides with the crank, and the perpendicular DF gives the value of t. If measured

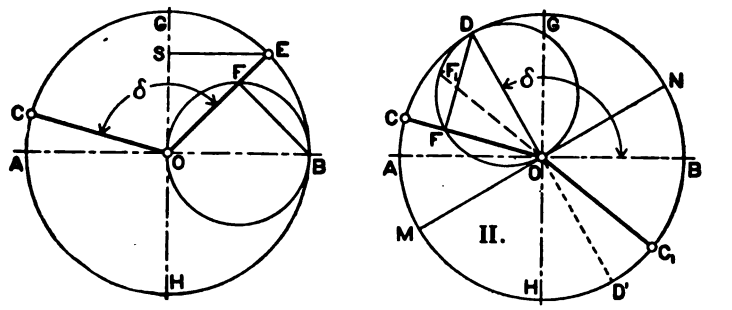

FIG. 222. — The Zeuner Diagram.

upward from MN, parallel to OP, t is toward the right, or plus; if downward, or in the direction OQ, it is toward the left, or minus. This is the Reuleaux or ordinate diagram of valve movement.

The derivation and form of the Zeuner or polar diagram are given in Fig. 222. We first develop, at I, a new way of representing the motion of the slide in terms of that of the driving crank arm (here the eccentric arm OE), as follows:

On the line OB, which is the right-hand dead-center position of the eccentric, draw the circle OFB with r or OE as its diameter. Then the intercept OF, cut from OE by this circle, is equal to t; for the right-angled triangles OBF, EOS, are always equal, hence also the sides OF and ES. Inspection of the figure, or a few trial constructions, will show that when the eccentric lies across this circle, so that the intercept OF is cut from it directly, the valve is to the right: but when the eccentric has to be produced back through O in order to cross the circle — that is, when it is anywhere in the semicircle HAG — the valve is to the left. Now, just as in Fig. 221, we change from a diagram in terms of

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eccentric position to one in terms of crank position, by rotating backward about O, through the angle δ, the figure made up of the circle OFB and the eccentric radius OE. Then, in II, the circle takes a constant position on the diameter OD, and the eccentric is brought into continual coincidence with the crank. The intercept OF cut from the crank by the valve circle measures t, to right or left as it is direct or indirect.

The geometry of both these diagrams is very simple: but practice and familiarity are needed to give facility in using and understanding them. A simple model, in which the crank-eccentric COE is made actually to rotate over either diagram, is a help at first in making it clear that the perpendicular DF in Fig. 221 or the intercept OF in Fig. 222 II, is always equal to ES. But after this has been clearly realized, the eccentric should be discarded, and the diagrams thought of only as showing a direct relation between crank angle and valve position. Not only do these diagrams give the length and direction of t, but we can see which way the valve is moving by noting whether t increases or diminishes as the crank advances, and can get an idea of the velocity of the valve by noting whether t is changing rapidly or slowly.

(*c*) RULES FOR DRAWING THE DIAGRAMS. — It is obvious that if the eccentric be placed on its right-hand dead center OB — when t will have its greatest plus value — the crank will be perpendicular to the base line MN of the Reuleaux diagram, and will lie along the valve-circle diameter of the Zeuner diagram. Then a rule for constructing the Reuleaux diagram would be: Draw an eccentric circle, of radius r; place the eccentric on its plus dead center and draw a diameter at right angles with the corresponding position of the crank: this will give the base line and determine the direction of $+t$. For the Zeuner diagram, place the eccentric on its plus dead center, and on the corresponding crank line measure off r and draw a valve circle on this radius as a diameter. These rules become general if we make the following assumptions: Let the initial dead center, from which to estimate crank angle, be that for which the piston is farthest from the crank shaft; let t be considered plus when it is from mid-position toward the shaft; and let the eccentric angle δ be always measured from crank toward eccentric, in the direction of rotation of the shaft:* then no matter which way the engine stands or runs, and whether or not the lines of piston stroke and valve stroke agree, the above rules, and the directional meanings of t as there stated, hold true.

* An exception to this statement of general conditions is found in the case of the locomotive engine as usually viewed — see Fig. 229.

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PROBLEMS

1. With values of δ near the middle of each of the four quadrants from 0 deg. to 360 deg., draw motion diagrams of both kinds, and show on each the crank positions where the valve is at mid-stroke and where t has its greatest plus and minus values. In some of these diagrams, take the engine conditions to be other than those of Fig. 162, as to position and as to direction of turning.

2. For given values of r and δ, draw a Reuleaux and a Zeuner diagram: and on each find where the crank is when $t = +\frac{1}{2}r$ and $t = -\frac{1}{2}r$.

(*d*) THE COMPLETE VALVE DIAGRAM. — Having established methods for completely representing and determining the movement of the valve, we shall next consider how this valve, moving back and forth over the ports, effects the steam distribution. In Fig. 223 I, a common slide valve is shown in mid-position on its seat; and the controlling dimensions, besides r and δ as represented at II, are

s = outside lap, or steam lap;
i = inside lap, or exhaust lap.

Complete valve diagrams for the left port, or for the head end of the cylinder, according to the two methods, are given at III and IV.

When the crank is at OM — in IV this line is tangent to the valve circle, so as to have a zero intercept — the valve is in mid-position. As the crank advances, the valve moves toward the right, the t ordinate being positive and increasing in both figures: when C gets to Q, where $t = Qq = cO = s$, the valve edge and port edge are just in line, or the port is just beginning to open. For the crank on dead center, the valve takes the position shown at V; the travel is t_0 and the port is open by the small amount e, which is called the lead. When the crank is at any position OC, to which VI corresponds, we have $t = CF = EO$: and it is evident that, in general, the port opening is equal to $(t - s)$. In order to make a graphical subtraction of s from t, we draw in III the lap line QR parallel to MN at the distance s; and in IV, draw the lap circle cKd, with s as radius. Then the segment QDR and the crescent cDd are identical diagrams of port opening. We see that admission begins — or, we " have admission " — at Q, maximum opening is at D, and cut-off takes place at R. It is evident that the determining of admission and cut-off is simply a matter of finding crank positions for which the valve is at a certain distance from mid-position.

After the crank passes R, the valve keeps on moving back from the right — as is shown by a plus but decreasing t — until it again gets to mid-position when the crank is at ON: then it goes toward the left, and soon opens the exhaust port, this occurring when $t = -i$. The beginning and end of exhaust, or " release " and " compression," as also

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the port opening during exhaust, are found by drawing the exhaust lap line TS or the inside lap circle fOe. In IV, instead of using only the plus valve circle on OD, we save overlapping by drawing another valve circle on OD′, for which the direct intercept shows left-hand or minus travel. This is convenient but not necessary, since it is evident that OT is determined equally well by either intersection, e or e′. In finding release and compression from the Zeuner diagram, the beginner is likely to confuse the intersections of valve circle and inside lap circle, especially when, as is usual, only the one valve circle is drawn. Keep clearly in mind, not only that the valve must be at a certain distance for one of these events, but also to which side it must be, and which way it must be moving. Thus, with the positive valve circle alone, if we were to draw a crank line from O through f for the release position, we should make a mistake: for while the valve is at the distance i, it is toward the right; whereas it should be to the left and moving to the left, as is the case when the crank is at eOT. For these short-lap measurements the Reuleaux diagram is clearer and more accurate than the Zeuner.

On the exhaust side of this valve there is over travel; for if we measure off the port width b, and draw VW parallel to ST, and the circle hkg at the distance b from the lap circle, we see that the valve travels more than enough fully to open the port. Sometimes there is a slight over travel on the steam side: but more frequently — and most of the time in single-valve gears with variable cut-off — the maximum opening for admission is much less than the width of the port.

For the other port, or the other end of the cylinder, the events and conditions are diametrically opposite to those shown, with a symmetrical valve: if the laps are not equal, they must be drawn in, and the required intersections found. Generally, both sets of lap lines should be drawn on a Reuleaux diagram, dotting those for the crank end. But in the Zeuner diagram, we usually draw but the one valve circle; and, with equal laps, the same intersections serve for both ports.

PROBLEM

3. For given values of r, δ, s, i, and b, draw a complete diagram by each method, showing on it the steam distribution — especially admission, cut-off, release, and compression — for both ends of the cylinder; and test for completeness of opening and for over travel on both steam and exhaust sides.

(*c*) VALVE AND PISTON DIAGRAMS. — Having established simple methods for finding the relation between the positions of the valve and of the crank, our next step is to extend these to the valve and piston. The primary, determining diagrams are shown in Fig. 224,

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where the valve diagram (of either form) is combined with the piston-position diagram from Fig. 163: then DC′ and CF are simultaneous determinations, to be used, respectively, as ordinate and abscissa in Fig. 225 — compare the similar combination in Fig. 169. The distortion from symmetrical steam distribution caused by the action of the connecting rod, notably the inequality in the cut-offs, is well brought out

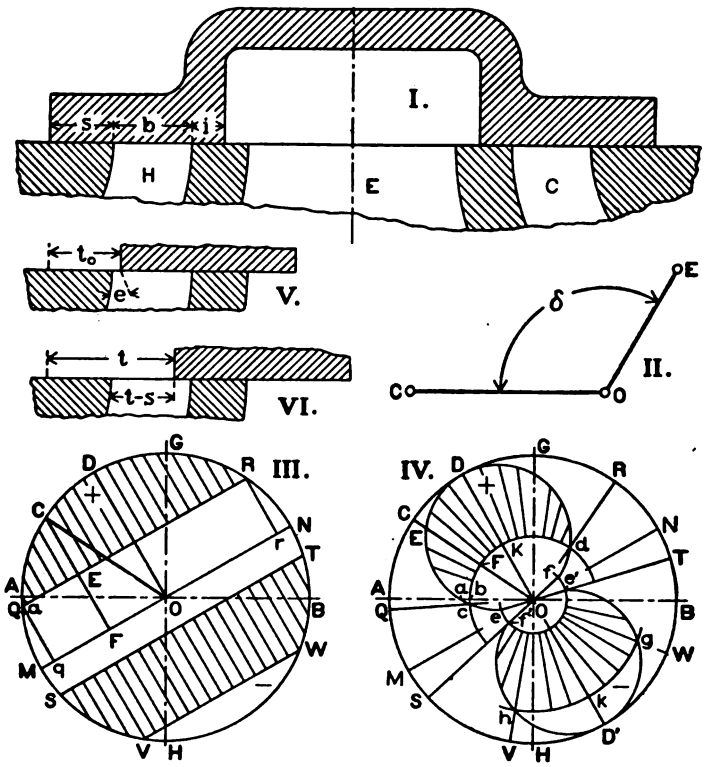

FIG. 223. — Valve Diagrams.

by this figure; but can be rather more clearly seen on the derived diagram given as Fig. 225, where the valve travel is plotted on the stroke line as a base.

The curve got by this method is elliptical in form, and with harmonic motion for the piston as well as the valve it is a true ellipse. The effect of the connecting rod is here shown by dotting in parts of the simpler curve. The lap lines are now drawn parallel to MN: and the four events, admission, cut-off, release, and compression, are located by the intersections marked A. B, C, and D, respectively. Dotted lines and primed letters are for the crank end of the cylinder.

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In order to equalize the cut-offs, making them the same as with harmonic piston movement, the laps would have to be changed to the values s_1 and s_2, as marked on the figure. This would reduce the lead

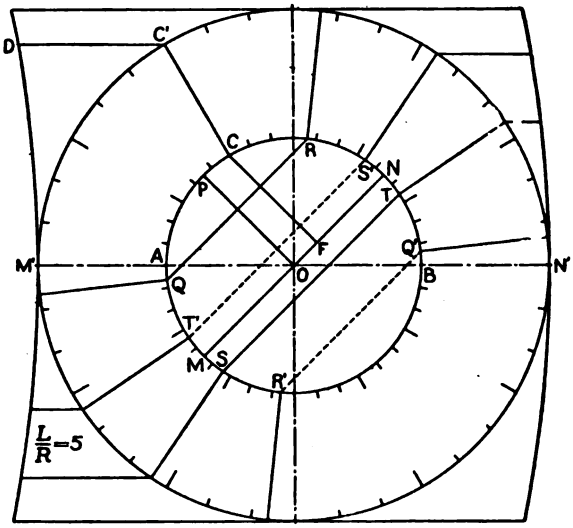

FIG. 224. — Combined Diagrams for Valve and Piston.

— which is shown by the distance from Q to the point where the curve is tangent to the end line — almost to zero at the head end, while nearly doubling it at the crank end, besides changing the widths of the port

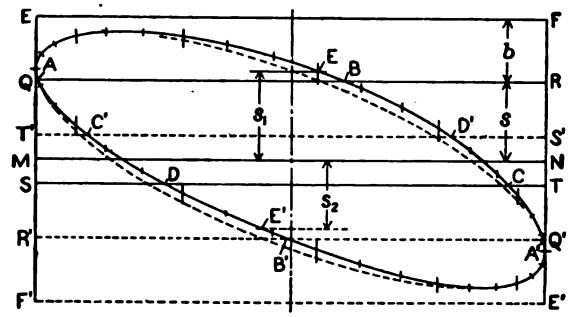

FIG. 225. — The Valve and Piston Diagram.

openings all through both admission periods. A diagram of this type furnishes the most satisfactory data for a comparison of valve action with realized steam distribution, as shown by the indicator; and this matter of symmetry of action will be more fully discussed farther on.

For an autographic diagram, to be drawn by the engine and to serve as a test of the working of the valve gear, especially with releasing

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gears of the Corliss and similar types, this elliptical curve is the most convenient, the motions necessary in the apparatus being the same as in the steam-engine indicator.

(*f*) LAP, LEAD, AND ANGLE OF ADVANCE. — The evolution of the engine valve is illustrated in Fig. 226. The simplest possible case is

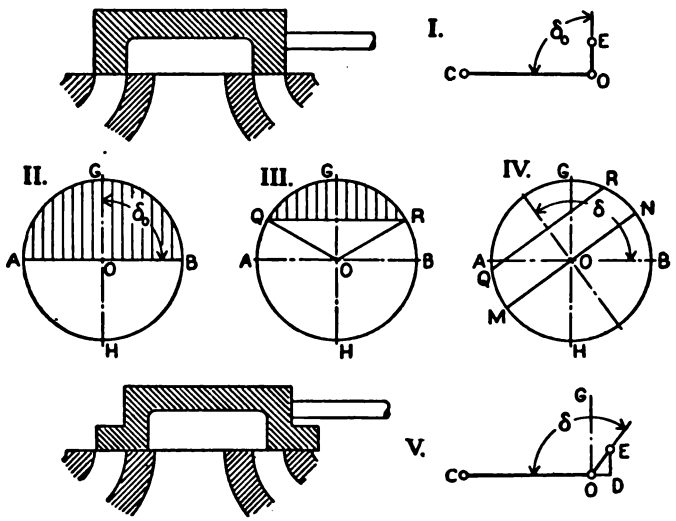

FIG. 226. — Evolution of the Valve.

shown at I, where the valve just covers the port when in mid-position, and is driven by an eccentric at right angles to the crank, so that its mid-position coincides with the dead center. This arrangement has the very decided fault that the admission is too long, the port being open during the entire half-revolution, or the whole stroke of the piston, as shown by the Reuleaux diagram at II.

To shorten the period of opening, the first step is to give the valve a lap, so that it will not uncover the port until the eccentric has turned through a certain angle from the vertical, and will close it at the same angular distance before the other mid-position, as in III. Along with this change, the eccentric must be advanced beyond the position at right angles to the crank, so that when the latter is on dead center the valve will be at a distance from its mid-position equal to the lap plus the lead. The effect upon the diagram is shown at IV; in V the valve is sketched, and what is often called the angle of advance, $\gamma = (\delta - \delta_0)$, or $(\delta - 90^\circ)$ in the usual engine, is determined by the relation

$$OD = s + e = r \sin (\delta - 90). \quad . \quad . \quad . \quad . \quad . \quad (204)$$

An important deduction from this figure is, that if the admission is

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to be very short, the lap must be very large relative to the radius, and the width of port opening correspondingly small. This matter, together with the changes in the action of the valve on the exhaust side, will be considered when we come to valve gears with a variable eccentric.

(*g*) TWO TYPES OF VALVES. — There are two typical forms of the slide valve, the flat and the piston form, and these are made with a great variety in detail. Examples of both have been shown in the last chapter, and a more detailed description will be found in § 42. Another distinction now to be drawn, and one having to do rather with the present side of the subject, is illustrated in Fig. 227. The first arrangement, having the live steam at the ends and the exhaust in the middle, as in the plain flat valve, is called the direct valve; the second,

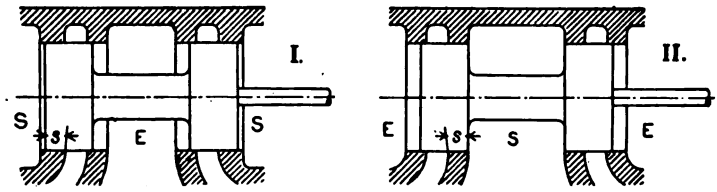

FIG. 227. — Direct and Indirect Valves.

with the steam in the middle and exhaust past the ends, and with the laps interchanged accordingly, is called indirect. Another way of defining the two types is to say that they have respectively outside and inside admission. And a general distinction, applying equally well to single, separate-function valves, is expressed by stating that the direct valve opens inward, moving toward the middle of the cylinder, the indirect valve opens outward — referring particularly to the steam edge at either end. In other words, the direct valve opens the port by moving in the direction of the piston stroke for which this opening is a preparation, while the indirect opens against the stroke and closes with it.

The effect upon the position of the eccentric, due to a change from the direct to the indirect valve, is shown in Fig. 228: the necessary reversal of each valve-travel distance is secured by reversing the eccentric in II into a position diametrically opposite to that which it occupies in I. These are the characteristic eccentric settings for the two types of valves. In II it is simpler to estimate δ as a negative angle, rather than to measure it all the way round in the plus direction.

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(*h*) EFFECT OF A REVERSING ROCKER ARM. — Sometimes a rocker arm pivoted at or near the middle is interposed between the eccentric and the valve. The result of this is brought out in Fig. 229, which is drawn for the locomotive, where the cylinder and axle interchange the

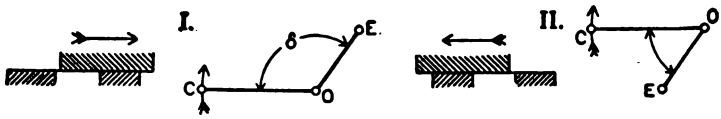

FIG. 228. — The Two Eccentric Settings.

characteristic positions for the stationary engine, as given in Fig. 162, but the zero dead center is still taken at the left. To compensate for the reversal of motion by the rocker arm, the eccentric must be diametrically reversed on the shaft; so that with a direct valve it has the setting proper to the indirect, and vice versa. In this case, however,

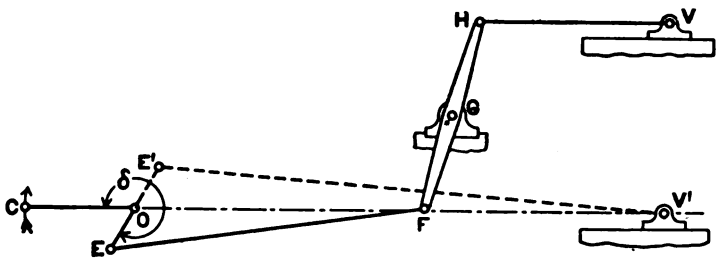

FIG. 229. — The Reversing Rocker Arm.

it is usual to disregard the reversal in the mechanism, and to draw the valve diagram as for an unreversed eccentric, in the usual position. It is evidently from its analogy to this effect upon the eccentric setting that the name " indirect valve " is derived.

§ 38. Various Valve-gear Relations

(*a*) THE BILGRAM DIAGRAM, while less simple in idea than either of those developed in the last section, is quite a little used, and the student should understand it. In Fig. 230, MN and OD are the same lines as in Figs. 221 and 222. If N is taken as a center or pole and a perpendicular NK is dropped upon any crank line, COK, the distance NK is the valve travel *t*: it is a simple matter to trace out the equality of the

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three triangles NKO, OFD, and CEO, thus showing that the Bilgram vector NK is the same as the Reuleaux ordinate CE and the Zeuner vector OF. To see whether the valve is to right or left, stand at O and look along the crank line toward C: if N is off toward the right from OC, the valve is to the right of mid-position. Lap circles are drawn with N as center, then tangent crank-line diameters locate admission at OQ, cut-off at OR, release at OT and compression at OS.

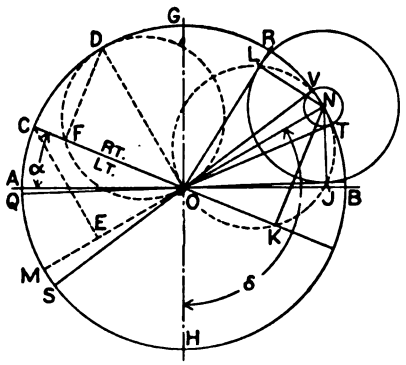

FIG. 230. — The Bilgram Diagram.

The dotted circle on ON is the locus of the point K; it is closely analogous to the Zeuner circle, and the vector is of the same character, but is measured from the outer end of the radius r or ON, instead of from the inner end. The following trigonometrical relations are easily traced out, α being always the crank angle AOC:

In Fig. 221, $\angle EOS = \angle DOF = (\alpha + \delta) - 90$,

then $ES = DF = t = r \sin EOS = r \cos(\alpha + \delta)$.

In Fig. 222, II, $\angle DOF = (180 - \delta) - \alpha = 180 - (\alpha + \delta)$,

then $OF = t = r \cos DOF = r \cos(\alpha + \delta)$.

In Fig. 230, $\angle NOK = (\delta - 90) + \alpha = (\alpha + \delta) - 90$,

then $NK = t = r \sin NOK = r \cos(\alpha + \delta)$.

(*b*) GEOMETRICAL RELATIONS. — Certain geometrical properties of the valve diagrams, which have frequent application in problems upon the working of the valve as represented by these diagrams, are illustrated in Fig. 231.

1. If the crank be placed on its zero dead center OA, the particular position OE of the eccentric radius and the line OD will be symmetrical with respect to the vertical GH. And if the crank be turned to OB, then OE′ and OD are symmetrical with respect to AB.

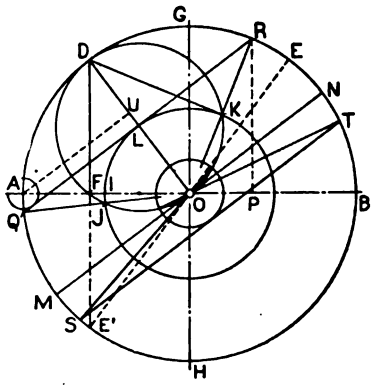

FIG. 231. — Geometrical Relations.

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2. By drawing the two diagrams together, the identity of their determinations is made apparent: as also the fact that a crank position dependent upon a short valve travel, as OS or OT, is much more accurately located by the Reuleaux diagram than by the polar. Even when using the Zeuner diagram alone, we make an accurate determination of S and T by drawing ST tangent to the lap circle and at right angles to OD.

3. A perpendicular from D upon AB cuts off a length OF equal to the steam lap plus the lead. Conversely, if we measure off $(s + e)$ and erect a perpendicular, this line is a locus of D. The complementary relation for the eccentric is shown in Fig. 226 V, and stated in Eq. (204): it is what determines the eccentric angle δ in practical valve setting.

4. The fact that DK is tangent to the lap circle is especially useful when we have a locus of D and wish to draw the valve diagram which will give a particular cut-off.

5. The line DO bisects the angle of admission QOR and the angle of release TOS.

6. A circle from A with the lead e as its radius is tangent to the line QR: self-evident on the Reuleaux diagram, this can be independently shown for the Zeuner from the equality of the right triangles AUO, DFO.

Any event in the valve action can be located by giving either the crank angle at which it takes place, or the corresponding piston travel with infinite connecting rod. Thus the cut-off is fixed either by the angle AOR or by the ratio of AP to AB: but the admission line OQ can be located only by the angle of lead, AOQ or ϵ.

(*c*) Problems on the Simple Valve Gear. — The following résumé of the symbols used in this discussion will be found convenient:

r = radius of eccentric, half of total travel of valve.
l = length of eccentric rod.
δ = eccentric angle, measured from crank toward eccentric in direction of rotation.
t = valve travel, or distance from mid-position at any instant.
s = steam lap, outside on a direct valve, inside on an indirect.
i = exhaust lap.
b = width of steam port.
e = lead, measured in port opening.
ϵ = angle of lead, plus when measured from admission line toward dead center.

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A few practical problems will now be given, all having a direct bearing on valve setting or design. Others can be devised, but many of them are useful only as illustrating the geometrical possibilities of the diagrams.

PROBLEMS

4. Given r, s, and e: find δ and cut-off.
5. Given r, ϵ, cut-off, release: find δ, laps, and compression.
6. Given r, e, cut-off, and compression: find δ and release.
7. Given s, e, and cut-off: find r and δ.

(*d*) VALVE SETTING. — The amount of adjustment possible after the engine has been designed and built varies with the type of valve gear. In many single-valve engines, where the eccentric is carried by a shaft governor or where it is keyed to the shaft, everything depends on its being correctly designed: but sometimes the eccentric can be rotated on the shaft, so as to change the angle δ, and clamped in any desired position. In most engines the length of the valve rod or of the eccentric rod can be varied; and in the more complex valve gears there are likely to be a number of points at which this kind of adjustment can be made.

The two conditions to be met are, first, that the valve movement shall be symmetrical, so that the steam distribution will be as nearly as possible the same for the two ends of the cylinder; second, that it shall be properly timed with reference to the motion of the piston.

With the engine cold and the valve chest open, the two adjustments, of rod length and of eccentric angle, would be made together until the leads were equal and had the proper value — the engine being repeatedly placed first on one dead center, then on the other. Or, if desired, equality of leads may be partly sacrificed to equality of cut-offs. With the help of the indicator, the valve can be set with the engine in running condition, stopping it for adjustment after each trial. This latter is, in many cases, the final method.

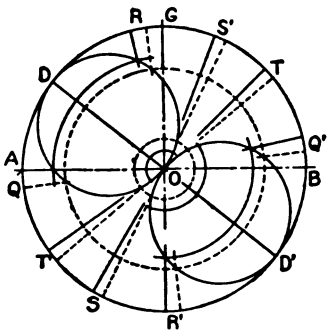

FIG. 232. — Rod Length Changed.

(*e*) CHANGE OF ROD LENGTH. — The effect of this adjustment is shown in Fig. 232: the dotted lines show symmetrical working or equal laps, the full lines the result of lengthening the rod, in a direct-valve engine; two circles being used so as to separate the indications for the two ends. Referring to Fig. 223 I, we see that to shift the mid-posi-

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tion to the left will increase s and i', decrease i and s' — this notation distinguishing the ends just as does that used for the events on the diagram. Then for the head end, admission is shortened and exhaust lengthened; while the opposite effects are produced in the other end.

Adjustment under the indicator is illustrated in Fig. 233. As shown by I, there was quite an inequality in the cut-off and in the power developed in the two cylinder ends: this could also be detected

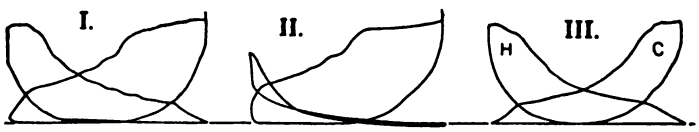

FIG. 233. — Valve Setting with the Indicator.

by the sound of the exhaust puffs. Through uncertainty as to the type of valve, the rod length was at first altered in the wrong direction, with the effect shown at II. Reversing this, and correcting in the proper direction, the symmetrical steam distribution represented at III was secured.

The matter of proper setting of the eccentric will be discussed in connection with the Corliss valve gear. In any case, a thorough understanding of the working of the mechanism is fundamental to an intelligent treatment of faults in its operation.

(*f*) SECONDARY DISTURBANCES. — Under this heading may be mentioned looseness of the joints of the gear and the angular swing of the eccentric rod. The former will cause the valve to make a little pause at each end of the stroke, and then lag behind its geometrical position by an amount equal to one-half of the combined play in the joints: this being true if friction of the valve is the predominant force, as against inertia, which would throw the valve outward from mid-position, or pressure of steam upon the end of the valve rod, which tends to keep the lost motion all taken up in one direction.

The influence of rod swing can be directly shown on the Reuleaux diagram, by substituting for the straight reference line MN in Fig. 221 an arc with the rod length l as radius — this being analogous to the curved tangent arcs in Fig. 163. The lap lines will then be " parallel " arcs of the same radius. This effect can commonly be disregarded; but a full discussion of it will be found in STEAM ENGINE, Vol. II, pages 202 to 206.

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§ 39. The Shifting Eccentric: Variable Steam Distribution

(*a*) MOVING THE ECCENTRIC CENTER. — Following the line of development suggested by Fig. 226 III and IV, and carried forward in Relation 3 under Fig. 231, we see that if the center of the eccentric be shifted along a line at right angles to the crank arm, changing both the length and the inclination of the eccentric radius, the cut-off will be varied without changing the lead. This fundamental principle of the whole class of single-valve, variable cut-off engines (as well as several derived forms) is illustrated in Fig. 234.

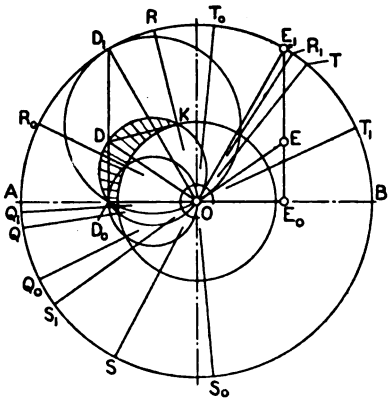

FIG. 234. — Shifting Eccentric with Constant Lead.

The eccentric is supposed to be carried on a cross slide keyed to the shaft, so that the center can be moved along the path E_1E_0; and is either clamped in any particular position, or held in place by the governor. For the longest radius OE_1, the valve circle is on OD_1; and all the events, cut-off at OR_1, release at OT_1, and exhaust closure at OS_1, are late.

The intermediate diagram is located so as to give cut-off at three-eighths of the stroke, by drawing KD perpendicular to the radius OR: and along with the change in cut-off go smaller changes in release and compression, all these events being made earlier by the increase of δ.

The limit of movement of the eccentric is usually at E_0, on the crank line: and the corresponding steam distribution is shown by the circle on OD_0. The very small opening of the port, together with the great compression from S_0, produces a steam diagram whose effective area is not far from zero.

(*b*) THE ECCENTRIC PENDULUM. — Referring to Fig. 15, where a simple shaft governor is outlined and described, we see that what needs to be known from it, in order to determine valve movement, is the shape of the locus or path of the center E on the plane of the governor — this locus being commonly an arc of a circle, traced by the end of a swinging bar or "pendulum," of which the position and displacement are determined by the equilibrium of the forces within the governor proper. For valve-gear purposes, the required dimensions are

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shown on Fig. 235, where the different possible arrangements are given and a conventional method is developed for stating, as concisely as possible, the essential data.

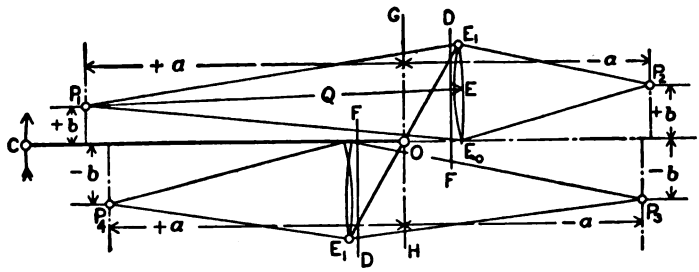

FIG. 235. — The Eccentric Pendulum.

With the crank line CO and the perpendicular GH as axes, the pivot P is located by the coördinates a and b, the algebraic signs having the meanings indicated. In general, for any position of the crank, a is plus from O toward C, b is plus in the direction of the motion arrow through C. Then the pendulum radius Q or PE, and the limiting eccentric radius r_1 or OE_1, complete the data. In the absence of specific statement, E_0 is supposed to be on the line CO.

Of the four characteristic positions of P, numbers 1 and 2 belong to the direct valve, 3 and 4 to the indirect. The offset b may have any value from zero up to r_1; making it equal to half the vertical projection of r_1 gives the nearest approximation to the straight-line locus of Fig. 234. In any case, drawing a line DF at a distance equal to the steam lap s from GH will show clearly how the lead will vary as E changes its position.

(c) DIAGRAMS FROM AN AUTOMATIC CUT-OFF ENGINE. — In the example worked out in Fig. 236, the data are: $a = +5\frac{1}{2}''$, $b = +1\frac{1}{2}''$, $Q = 6\frac{3}{4}''$, $r_1 = 2\frac{5}{8}''$, $s = 1''$, $i = 0$. The eccentric locus, from E_1 to the crank line, is divided into three equal parts, and four Reuleaux diagrams are drawn, a part of the lap circle helping to locate the several lap lines. The change in admission and cut-off is shown by drawing in the corresponding crank positions, while release and compression would be located by simply extending the base lines of the diagrams.

The curves in Fig. 237 are plotted from the diagrams in Fig. 236. The four distances QE, Q'E', NH, MH', are all equal to the port width b, which is $2''$: and even with the full stroke of the valve the port is not fully opened for admission, though there is over-travel on the exhaust side in this one case. Curve No. 3 gives only a moderately early cut-off, and yet we see how small is the port opening, and how

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great an effect can be produced by even a slight change in the length of the valve rod. Note that in case 4, where the eccentric is in line with the crank, the motion diagram reduces to a single line, which would be straight with the piston in harmonic motion, but is here slightly curved.

(*d*) PROBLEMS ON THE SHAFT GOVERNOR. — The following data are from actual engines, as were those for Fig. 236. In each problem, get the eccentric locus first, and then draw valve diagrams for the greatest

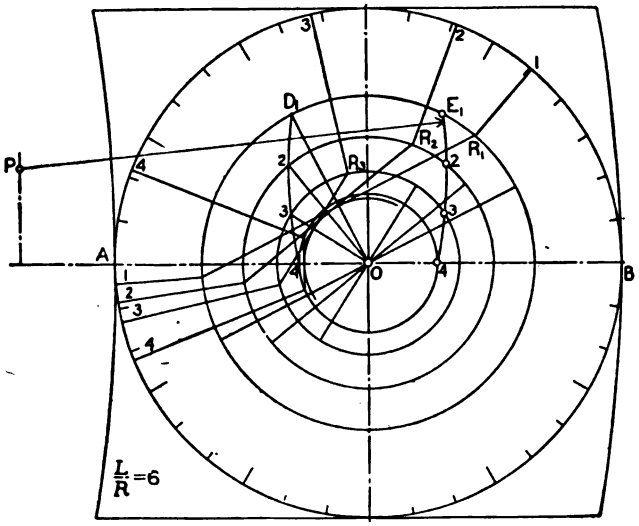

FIG. 236. — Valve Diagrams from a Shaft Governor.

eccentric radius, for cut-off at one-third of the stroke, and for the earliest cut-off. One-third is chosen because it will give an effective cut-off, referred to the boiler pressure, at about one-quarter of the stroke; and it is upon this cut-off that the rated power of the engine is based. The Zeuner diagram is rather better for illustration, the Reuleaux for accurate determination of the whole movement.

PROBLEMS

8. Direct valve, $a = +14''$, $b = 0$, $Q = 15''$, $r_1 = 1\frac{3}{4}''$, $s = 1\frac{3}{16}''$, $i = 0$.
9. Indirect valve, $a = -1\frac{1}{4}''$, $b = -\frac{7}{8}''$, $Q = 2\frac{3}{4}''$, $r_1 = 1\frac{7}{8}''$, $s = 1\frac{1}{16}''$, $i = \frac{1}{8}''$.
10. Direct valve, $a = +5\frac{1}{4}''$, $b = +\frac{3}{4}''$, $Q = 6\frac{1}{4}''$, $r_1 = 1\frac{5}{8}''$ $s = 1''$, $i = \frac{1}{4}''$.

(*e*) WIDTH OF PORT OPENING. — Ideal valve action would be characterized by a very quick (practically "instantaneous") movement in opening and closing the port, together with a very full width of

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opening. Figure 237 shows how far these requirements fail of fulfilment, especially when the cut-off is early. Let us suppose that in this particular engine the width of the port is proportioned so as to give a certain maximum velocity of steam flow, say 200 ft. per sec. — see § 42 (e) for the method of determining this width. The idea is that when the piston has its highest velocity, practically equal to the constant v_0 of the crank pin, the current of steam flowing through the full port at 200 ft. per sec. will just fill the volume being displaced by the piston. In other parts of the stroke, the flow along the port or passage will be slower: but in the curves V_1 and V_2 is represented the variable width

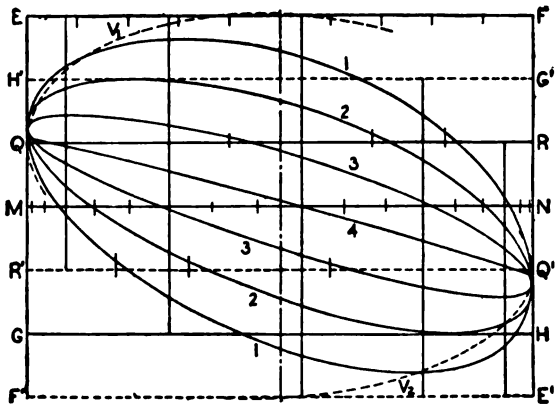

FIG. 237. — Stroke-line Diagrams from Fig. 236.

of opening, by the valve, that corresponds to a maintained speed of 200 ft. per sec. for the steam. To get an ordinate at any piston position, multiply the port width by the ratio of actual piston velocity v (as in Fig. 166) to crank-pin velocity v_0: these widths are measured from the lap lines or port edges QR and Q'R' as bases. From a comparison between the dotted and the full-line, actual curves, it is very evident that the actual velocity of flow past the valve must greatly exceed the assumed value of 200 ft. per sec., especially at early cut-off, with consequent large drop in pressure toward cut-off. Another point is, that the opening ought to be a little greater at head end than at crank end, on account of the somewhat higher velocity of the piston in the first part of the forward stroke.

(*f*) SYMMETRICAL ADMISSION. — In Fig. 237, the mean or equal cut-offs for curves 1, 2, and 3 are marked by short vertical lines: as in Fig. 225, it appears that in order to locate the act of valve closure at the same relative position in both strokes, it will be necessary to use a

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larger steam lap at the head-end port than at the crank end. But equality in effective cut-off, as defined in § 19 (*d*), or in the amount of steam admitted to the respective ends of the cylinder, is far more important than to have the valve close at the same fraction of both strokes. Considering the higher piston velocity in the forward stroke, a very small unbalancing of the laps, in the direction just indicated, will be enough to secure equality in m.e.p.: its amount can be determined only with the help of the indicator, the adjustment being a simple matter of change of rod length, as in § 38 (*e*).

(*g*) INDICATOR DIAGRAMS. — To illustrate variable steam distribution by the slide valve, the steam diagrams in Fig. 238 have been made

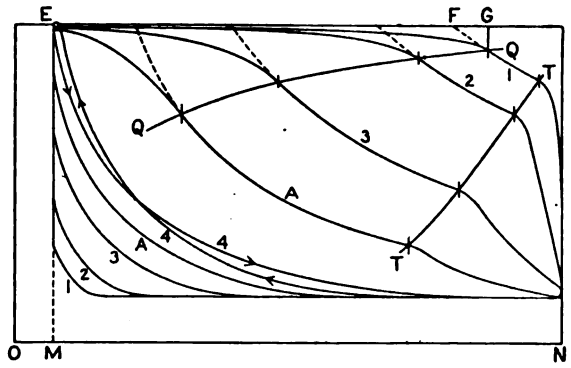

FIG. 238. — The Variable Steam Diagram.

up. They correspond with the timing of events in Fig. 237, but are laid out in close imitation of actual indicator cards. The extra diagram A is drawn for valve closure at one-fourth of the stroke, which, however, makes the effective cut-off a good deal earlier. The curves are, of course, smoother than those traced by the indicator, especially with high-speed engines.

The most important things shown in Fig. 238 are the locus QQ of valve closure as related to pressure, the locus TT of the point of release, and the manner in which compression changes with cut-off — all of them characteristic of the type of engine under consideration. Curve QQ has a large share in determining the relation of power (or of m.e.p.) to valve action and, back of that, to governor position. Its general form is rational, but any attempt at a close preliminary layout can be made only by applying experience with other engines of similar design.

The realized performance of any shifting-eccentric valve gear can be clearly shown by a combination of diagrams like that in Fig. 238,

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with perhaps some derived curves representing the variation of m.e.p. with cut-off, or of cut-off, compression, and m.e.p. with the position of the eccentric center on its locus.

The influence of speed of running, in modifying the steam distribution effected by a given valve action, has been well illustrated in Fig. 99.

(*h*) INFLUENCE AND VARIATION OF LEAD. — The matter of proportioning this element of the valve action so as to get the best results is one that can hardly be reduced to definite terms. In general, two objects are to be kept in view: first, to have the engine run smoothly, without a too abrupt reversal of pressure at the end of the stroke; second, not to waste any of the possible area of the steam diagram by late admission. As regards the first, nothing need be added to the brief statement at the end of § 34 (*c*), except to say that a " negative " lead may be called for under certain conditions. As to the matter of

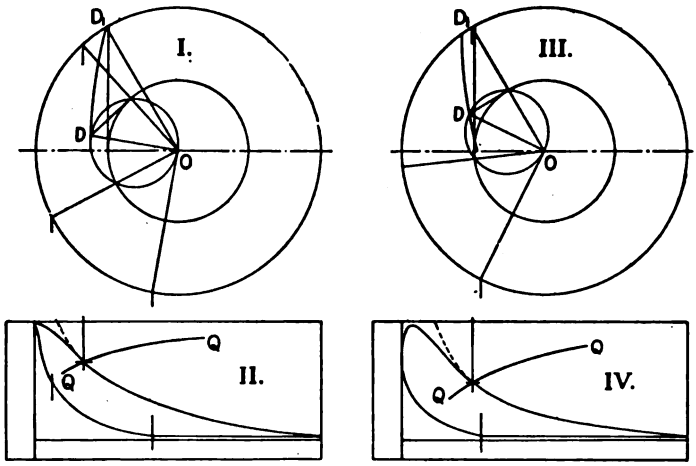

FIG. 239. — Different Ways of Varying the Lead.

getting full steam pressure before the piston begins its stroke, it is evident that rotary speed, clearance volume, and height of compression all exert influence. In any case, the angle of lead — a measure of time — is rather more important than the exact width of opening at the instant of passing dead center. In an engine where the admission is controlled by a fixed eccentric, this angle will usually lie between 5 and 10 deg.

Engines of the shifting-eccentric type, with the shaft governor, show considerable variety in the location of the pivot point P, Fig. 235, and in the manner of variation of the lead. As typical cases in this respect, consider the diagrams sketched in Fig. 239. The point P is on the

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same side of the center in both I and III; but in the first case it lies on the crank line, in the second it is on a parallel line through D_1. The valve diagram in I is drawn for cut-off at one-sixth of the stroke; and through the large lead quite a wide opening of the port is secured, even with this very early cut-off. In III and IV, the mechanical cut-off, or instant of complete closure, is at one-quarter stroke: the greater drop in pressure, due to throttling on account of the small opening, is clearly shown, especially by the lower position of the curve QQ.

One advantage of the arrangement at III is that the governor can completely shut off steam from the engine, which it cannot do with the proportions in I. Further, the eccentric moves through a somewhat smaller distance, for a given range of power, in III than in I. It is suggested by the steam diagrams, II and IV, that the question as to the best method of governing the engine under small loads is here involved — this question being, whether it is better to throttle the steam or to make the cut-off very early. In view of what has been shown in §§ 22 and 23 as to the harmful effect of excessive expansion and compression, it is likely that a light-load diagram of the type at IV may give better economy than that at II.

§ 40. Reversing Valve Gears

(*a*) THE STEPHENSON LINK MOTION. — The derivation of this most common type of reversing valve gear is at once suggested by consideration of the eccentric settings for opposite directions of rotation, shown in Fig. 240. If the two eccentrics are put on the engine shaft side by

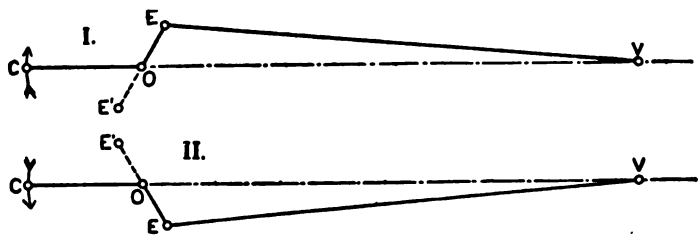

FIG. 240. — Eccentric Settings for both Directions.

side, and if the control of the valve can be transferred from one to the other at will, the engine can be made to run in either direction. What was logically the primary type of a device for making this transfer is that shown in Fig. 241, where by moving the lever R either hook, B_1 or B_2, can be made to engage the pin at V. This crude arrangement preceded the link motion, and was used on hoisting engines: it has, of course, been entirely superseded.

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The substitution of the curved link for the " gabs " of Fig. 241 is a radical improvement, not only on account of its mechanical superiority, but because it adds a second very important function to that of merely

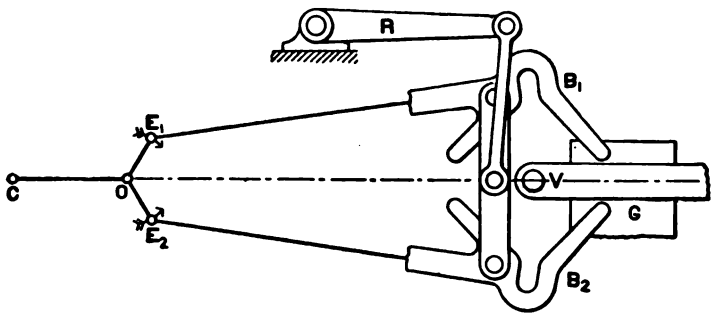

FIG. 241. — The Gab Motion.

reversing the engine, in making possible a regulation and variation of the steam distribution very similar to that effected by the movable eccentric of the shaft governor.

(*b*) FORM OF THE LINK MOTION. — This valve gear is used on locomotives, marine engines, hoisting engines, and rolling-mill engines. The example in Fig. 242 has quite an extensive transmission gear between the link and the valve, the rod 6, bent to clear an axle, running forward to a rocker such as is outlined in Fig. 229 and in Fig. 244. In the other lines of service named, the valve rod is generally driven directly by the block, giving the simpler scheme in Fig. 243.

The valve gear as a whole divides itself into several component parts. The link motion proper, from the shaft or main axle to the link 4, including the suspension rod 13, forms one complete mechanism, with definite motion; the adjusting gear, pieces 14 to 16, constitutes another division; the valve connections, comprising everything from the block 5 to the valve, make up a third. The link in Fig. 242 is of the offset type; that is, the pin connections for the eccentric rods are set well back of the center arc of the link. The hanger pin projects backward from a saddle which is bolted to the link and is so formed as not to interfere with the passage of the block. The block pin is fast on rod 6 (on the rocker in Fig. 244), and has a bearing in the block. In marine engines the rod pins are commonly on the center line of the link, by which arrangement some minor disturbances of movement are eliminated: the block slides between two curved bars, which are at its sides instead of its front and back as in Fig. 242, while the eccentric rods are forked so as to engage joint pins on the outside of these bars; and thus the block can get clear out into line with either rod.

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The reverse-lever system, or the adjusting gear, controls the power of the engine, by raising and lowering the link so as to change the position of the block in the link, and thereby vary the steam distribution. By means of a latch on the reverse lever, engaging notches on the

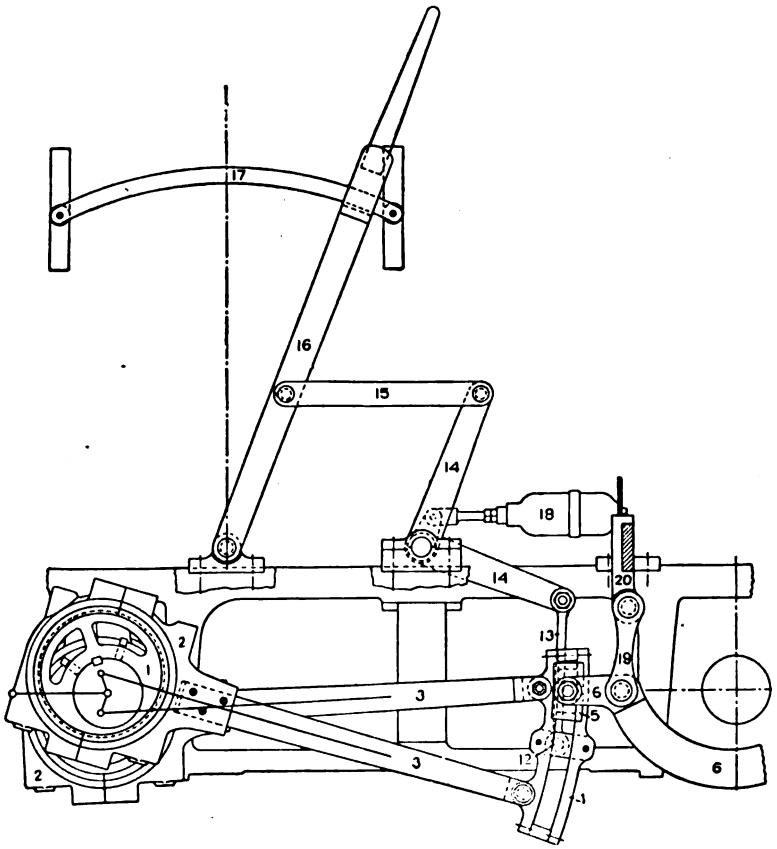

FIG. 242. — A Locomotive Valve Gear.

1. Eccentric.
2. Eccentric Strap.
3. Eccentric Rod.
4. Link.
5. Block.
6. Extension Rod.
12. Hanger Pin.
13. Suspension Rod.
14. Lifting Shaft.
15. Reach Rod.
16. Reverse Lever.
17. Notched Arc.
18. Balance Spring.
19. Extension-rod Hanger.
20. Bearing for Hanger.

arc 17, the valve gear can be locked at any setting. In the casing numbered 18 is a compression spring, which balances the weight of the link motion, so that the reverse lever will move easily in either direction. The radius of the link arc is equal to the length of the eccentric

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rods; if then the axes of the two eccentric straps could be brought to the axis of the shaft and held there, the link could be moved back and forth without changing the position of the valve — except as the tipping of an offset link will change the effective distance (parallel to the valve stroke) between an arc through the rod pins and the center arc of the slot. This particular degree of curvature therefore reduces practically to zero any effect which the shape of the link might have upon valve position, and leaves the mechanism free to reproduce and combine, at the block, the motion of the eccentric centers.

(*c*) ARRANGEMENT OF THE RODS. — The two typical general forms of the link motion, differentiated by the manner in which the block drives the valve, are outlined in Figs 243 and 244 — both turned about

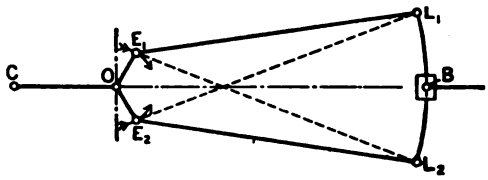

FIG. 243. — The Direct Link Motion.

into the characteristic locomotive position. The effect of the reversing rocker arm, as set forth in Fig. 229, and further indicated by the dotted-line eccentric settings in Fig. 240, is here shown in the reversed position of the crank, with reference to the figure of the eccentrics, in Fig. 244. These diagrams are also intended to make clear the distinction between the two possible arrangements of the eccentric rods, or two ways of connecting the eccentrics to the link — that between "open" and "crossed" rods. During a whole revolution of the shaft, the two rods will be part of the time clear of each other, as here, part of the time crossed, as in Fig. 242. The distinction is made by turning the shaft so that the two eccentric centers E_1 and E_2 are toward the link, or between the link and a vertical center line through O; and then noting, for this characteristic position, whether the rods are open, as shown by the full lines, or crossed, according to the dotted lines.

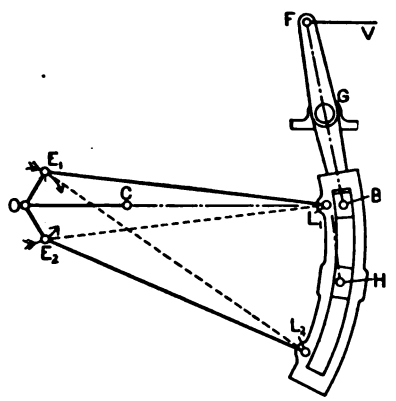

FIG. 244. — The Indirect Link Motion.

(*d*) MOVEMENT OF THE VALVE. — The following method of repre-

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senting the action of the link motion is not quite exact, because some small secondary movements of the link are not taken into account. Further, the proof of the statements as to proportionality along the eccentric locus and as to the shape of that locus is too long to be given here. A full discussion of the kinematics of the link motion, including the modified Gooch and Allen gears, will be found in STEAM ENGINE, Vol. II, pages 220 to 241. The approximate method for the Stephenson gear, quite good enough for all ordinary purposes, will now be set forth.

For any position of the block in the link, intermediate between L_1 and L_2, the motion of the valve is very nearly what would be given by a single eccentric with its center on a curve through E_1 and E_2: the location of this center E between E_1 and E_2 being similar — in the geometrical sense of proportionality — to that of B between L_1 and L_2. To make this statement fit Fig. 244 exactly, we should designate as B_1 and B_2 the block positions right in line with the two eccentric rods, and make the proportionality apply to the location of B on B_1B_2.

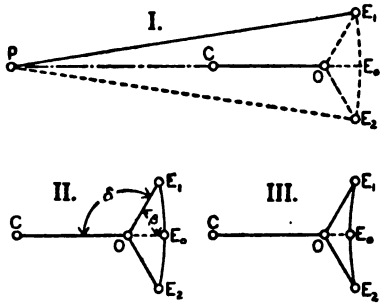

FIG. 245. — The Eccentric Locus.

In Fig. 245 the first thing shown, at I, is the manner in which reversing could be secured by the movement of an actual single eccentric center. Having an eccentric pendulum pivoted at P, extend its range of movement past E_0 to E_2; then the lower half of the locus of E will correspond with, and will produce, reversed running of the engine.

The locus of the equivalent single eccentric, for the link motion is shown at II and III of Fig. 245 for the respective cases of open and crossed rods. In strict accuracy parabolas, these curves are well enough approximated by circular arcs through the points E_1, E_0, and E_2: the point E_0 being located, or the length of the mid-gear radius OE_0 given, by the formula

$$r_0 = r\left(\cos\beta \pm \frac{k}{l}\sin\beta\right); \quad . \quad . \quad . \quad . \quad . \quad . \quad (205)$$

where r is the actual eccentric radius, β is the supplement of δ as on Fig. 245 II, k is half the length L_1L_2 of the link, and l is the length of the eccentric rod. The plus sign is for open rods, giving the convex locus in II, or increased lead toward mid-gear; the minus sign and the concave curve are for crossed rods. Note that the second term of the formula, $(k/l)r \sin \beta$, is the amount by which the curve departs from

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the straight line E_1E_2; being the fraction k/l of the half-length of this line.

The " adjustment " of the valve gear, or the position of the link upon the block, is defined as follows:

When the block is under the full control of either eccentric — that is, at L_1 or L_2 in Fig. 243, in line with L_1 or L_2 in Fig. 244 — the mechanism is said to be at " full gear," forward or backward. Mid-position is called mid-gear, as indicated above. Any other position is described by giving the fraction got by dividing the half-length of the link into the distance of the block from the middle of the link: thus we speak of half gear, three-quarters gear, and so on.

Positions of the reverse-lever latch on its arc will be almost exactly similar to those of the block in the link: so that the setting of the valve gear can be equally well defined in terms of that position.

(*e*) VALVE AND INDICATOR DIAGRAMS. — A sample set of valve diagrams for a Stephenson gear is given in Fig. 246. The eccentric

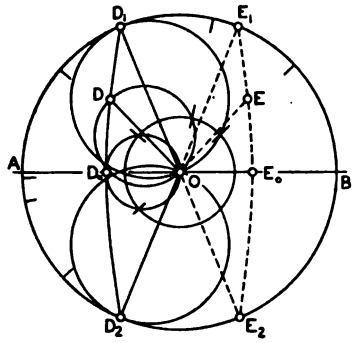

FIG. 246. — Valve Diagrams for the Link Motion.

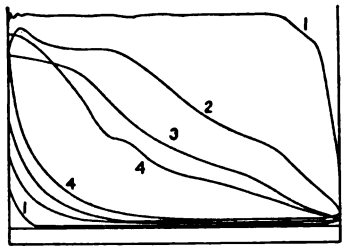

FIG. 247. — Diagrams from a Locomotive.

locus is first laid out at $E_1E_0E_2$ (for the crank at zero), and the locus of D is made symmetrical to it, with reference to the line GH. Valve circles are drawn, for full gear forward, on OD_1; for full gear backward, on OD_2; for mid gear, on OD_0; and for half gear forward, on OD; OD and OE corresponding with a position of block in link halfway between L_1 and the middle, according to the definition just given.

Several indicator diagrams from a locomotive are reproduced in Fig. 247. The largest, No. 1, shows the greatest amount of work per revolution that the engine is capable of performing; this would be done only at very low speeds, and the tractive force, or tangential force at the driving wheels, due to this steam diagram, would be so great that it would slip the wheels if friction were not increased by the use of sand

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on the rails. The normal resistance of the train is represented by the smaller diagrams, taken at speeds of 35 to 75 miles per hour. Slight variations in boiler pressure and in the opening of the throttle valve, together with the influence of speed, account for the differences in the admission pressure, and render impossible the close determination of a cut-off locus like QQ on Fig. 238.

PROBLEM

11. Given r, s, k, and l, and that the rods are open. First set the eccentrics so as to have zero lead in full gear; then get the eccentric locus, and draw Zeuner diagrams for full gear both ways, for mid-gear, and for quarter cut-off forward. Show the steam distribution in each case, taking the inside lap to be zero. Good average data are, $r = 2\frac{1}{2}''$, $s = 1''$, $k = 8''$, $l = 50''$.

(*f*) IDEA OF THE RADIAL GEARS. — The general principle of all gears in which a slide valve with harmonic motion from an eccentric is made to produce variable steam distribution is, to shift the center of the eccentric along a locus or path in the plane of the crank,* which locus approximates a straight line and lies at right angles to the crank line. In the shaft governor, the actual, physical eccentric is thus shifted. In the link motion, the variable eccentric is a geometrical resultant, got by proportional division between two actual eccentrics which form the ends of the locus. In a third scheme, now to be described, the effective eccentric is the resultant of two radial driving arms at right angles to each other, one of them variable in length.

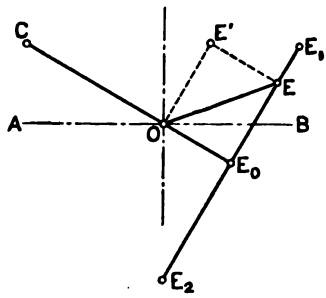

FIG. 248. — Component and Resultant Eccentrics.

The diagram in Fig. 248 shows the fundamental principle of what are called radial valve gears. To secure the desideratum of a driving center E which can shift from E_1 to E_2, use is made of a fixed arm OE_0 and a variable arm OE′, the point E′ changing position on a line parallel to E_1E_2. A mechanism is devised which will first produce a reciprocating motion from the eccentric OE_0, and then add to this the motion from OE′; the combined effect will be that of the resultant arm OE. The center E_0 is almost never an actual driving point, at the shaft, but the movement that would be given by OE_0 is derived from the main crank OC (generally from the crosshead) through a system of levers:

* That is, a plane perpendicular to the axis of the shaft, on which crank, eccentric, etc., are projected, and which turns with the shaft.

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for the OE′ component there may or may not be an actual eccentric at the shaft. A couple of examples will illustrate the working out of the idea.

(g) THE WALSCHAERT GEAR. — In Europe this is the most commonly used valve gear for locomotives, and it has been coming into increasing use in this country. The outline in Fig. 249 will serve as a

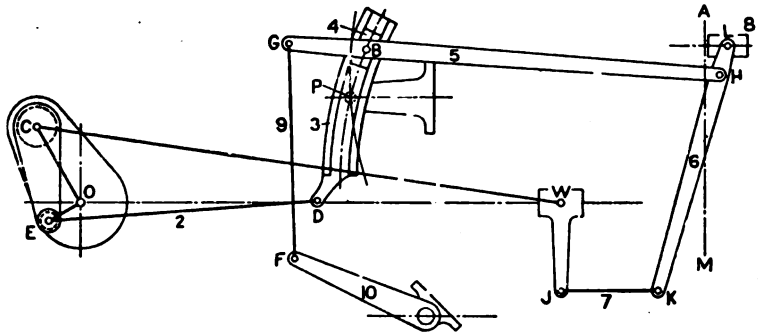

FIG. 249. — Outline of Walschaert Gear.

sufficient illustration. Motion taken from the crosshead at J is reduced and reversed by the lever KHL, taking H as the fulcrum: this gives the effect of a short arm opposite to the crank, like OE_0 in Fig. 248. The pin E, carried on a return crank from the main crank pin

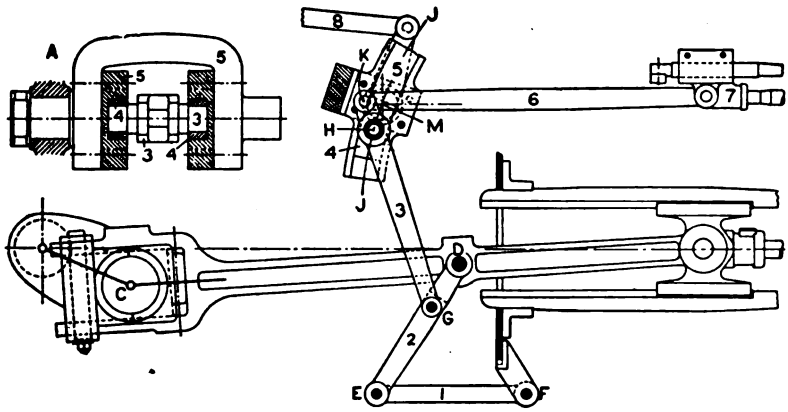

FIG. 250. — The Joy Valve Gear, as for a locomotive; at A, enlarged section of curved guide 5.

and having OE as its radius arm, drives the oscillating "link" 3: the resulting movement, communicated through block 4 and rod 5 to the pin H, is made variable by shifting the block in and out from the fixed

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pivot P, and reversible by carrying B past P. The reverse lever acts on the lifting shaft 10, just as in the Stephenson gear, Fig. 242.

(*h*) GEOMETRICAL RELATIONS. — At I in Fig. 251, B_0D_0 is the middle position of the link, with crank on dead center, while BD is the same as in Fig. 249. Let eccentric radius OE = r, DP = k_0, BP = k, also let maximum BP upward = k_1 and maximum BP downward = k_2. The displacement B_0B would come directly from an eccentric opposite to OE′ in III, and of the length $r' = r \times (k/k_0)$. At II, K_0L_0 is the middle position of the lever 6, KL_1 shows it as if swung on a fixed pin

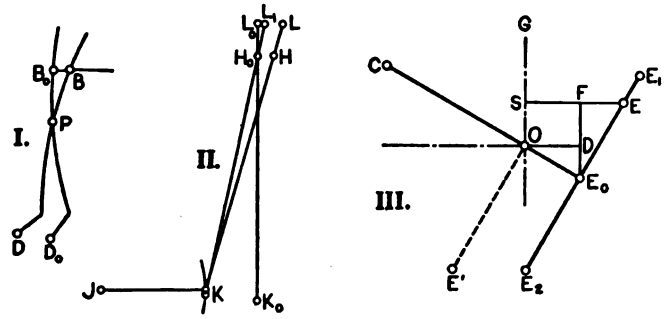

FIG. 251. — Derivation of Equivalent Eccentric, Walschaert Gear.

at H_0, and KL is the final position when H has been given the displacement $H_0H = B_0B$. For the lever 6, let

$$\frac{LH}{HK} = m, \qquad \frac{LK}{HK} = (1 + m);$$

then the displacement L_0L_1 is equivalent to that from an eccentric OE_0 (at III) of the length $r_0 = mR$. The distance L_1L is larger than H_0H in the ratio $(1 + m)$, so that the component arm E_0E has the value $(1 + m)\, r\, (k/k_0)$. At the outer limits of block position,

$$E_0E_1 = (1 + m)\, r \frac{k_1}{k_0}, \qquad E_0E_2 = (1 + m)\, r \frac{k_2}{k_0}. \quad . \quad (206)$$

In the diagram III, amplified from Fig. 248, SF is L_0L_1, FE is L_1L; the proportions are not relatively the same as in I and II, because the crank is now in a different position from that in Fig. 249.

PROBLEM

12. In an engine of 24″ stroke, the valve is to have a maximum travel of 5″, and lap plus lead or OE_0 is to be 1″: with r = 4″ and k_0 = 14″ in Fig. 249, find the ratio m for the lever LHK and the lengths k_1 and k_2 of the halves of the link.

(*i*) THE JOY VALVE GEAR. — In this mechanism there is no actual eccentric at all, but both of the component effects in Fig. 248 are de-

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rived from the main crank, through the connecting rod. The floating lever DE or 2 in Fig. 250 has one end pivoted on the connecting rod, the other guided by the radius rod 1. Lever 3, driver by the pin G, oscillates with a motion nearly proportional to that of the main cross-head, and moves bodily up and down because of the swing of the connecting rod. The first motion realizes the effect of the component eccentric OE_0, the second, through the slant of the guide or link JJ (which can be changed and reversed), is turned into the direction of the valve movement, and becomes equivalent in result to a variable driving radius E_0E.

§ 41. The Double-valve Gear

(*a*) General Considerations. — It has been made clear by Figs. 226, 234, and 236 that when early cut-off is secured with a single slide valve, the lap must be relatively very large and the port opening small; and that the compression must begin early in the return stroke, on account of the great angular advance of the eccentric. To prevent early compression from running too high, the clearance must be large; but while large clearance is incidental to the design, and unavoidable, in most high-speed engines, and while a good compression is mechanically advantageous when the inertia force of the reciprocating parts is large, a closer search for economy renders desirable a valve action which will not cut away so much of the possible effective area of the steam diagram as is lost, for instance, in Fig. 74. Simpler than the arrangements with separate valves and ports for admission and exhaust, and with no limitations as to speed, the Meyer or double slide-valve gear is the first type which we shall consider as meeting the requirements of full port opening, quick cut-off, and the most favorable regulation of the exhaust operations.

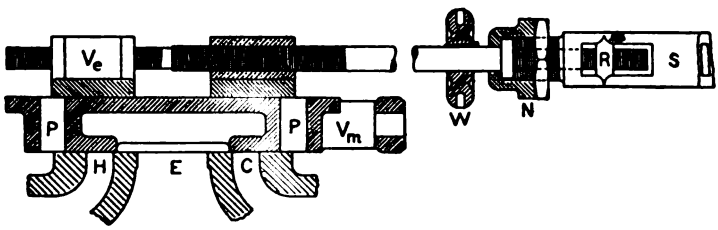

Fig. 252. — Meyer Valve with Variable Lap.

(*b*) The Meyer Valve Gear. — In Fig. 252 the main valve V_m is extended beyond the lap edges, so as to enclose the ports P, P; and on its flattened back rides the cut-off or expansion valve V_e, which controls the passage of steam through the ports in the main valve.

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The latter determines the operations of admission, release, and compression, just like a single valve: it is driven by a fixed eccentric, and does not cut off until late in the stroke, so that it can give a good, full port opening. Early and variable cut-off is effected by the riding valve. Very frequently the latter is driven by a shaft governor; but in Fig. 252 is shown an adjustable device (not automatic), often used on air compressors, where the governor acts by throttling, and both eccentrics are fixed. By means of right and left threads on the valve rod, the two parts of the expansion valve can be separated or brought together so as to make the cut-off earlier or later. When adjusting, the clamping nut N is slacked off a little and the rod turned by the spanner wheel W until the indicator R is at the desired cut-off, as graduated on the slide S: then the nuts are screwed up tight.

(*c*) RELATIVE MOVEMENT OF THE CUT-OFF VALVE. — In the diagram of the eccentrics, Fig. 253, E_m is the center that drives the main valve, while E_e drives the expansion valve; and the absolute movement of the latter, or its movement with reference to the fixed valve seat, is determined by the rotation of the eccentric radius OE_e. We are interested, however, not in the absolute position of V_e at any instant, but in its position on V_m: and this relative travel is got by thinking of V_e as driven on V_m by the eccentric radius E_mE_e, or r_v. The geometrical relation involved is simply that SE_e, the horizontal projection of E_mE_e, gives always the distance of the center line of V_e from that of V_m. The fact that this crank arm rotates about a moving point E_m is immaterial; but this movement of the center of rotation can be eliminated by shifting r_v to OE_v, or to the opposite side of a parallelogram of eccentrics: then since $S'E_v$ is always the same as SE_e, we have that V_e would be driven on a fixed valve seat by OE_v just as it is actually driven on its moving seat by E_mE_e. The radius r_v we call the virtual eccentric; and its angle δ is measured off as indicated on Fig. 253. It will be noted that the two valves are never simultaneously in mid-position, as they are drawn for illustrative purposes in Fig. 252.

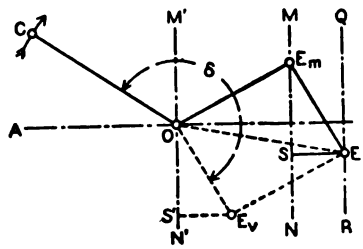

FIG. 253. — Relations of the Eccentrics.

(*d*) FUNCTIONS OF THE CUT-OFF VALVE. — It is required of this valve that, being out of the way when main-valve admission begins, so as to leave a clear passage into the port P, it shall then close this port quickly at some desired instant, earlier or later, in the stroke of the piston. Any crank-driven slide moves most rapidly when near the

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middle of its stroke; consequently, the cut-off valve must have a small lap if it is to be in rapid motion when closing its port. Very often the lap is negative, or the valve does not cover the ports when in mid-position, as is the case in Fig. 252.

(*e*) Positive and Negative Valves. — Certain ideas fundamental to the whole matter of valve action are given concrete and complete expression in Fig. 254. By means of primary diagrams of the ordinate

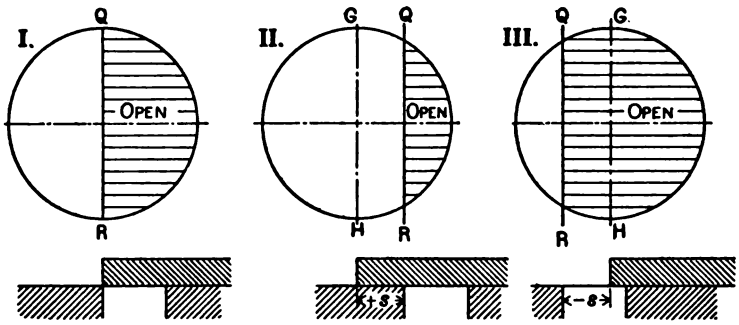

Fig. 254. — Effect of Varying the Lap.

type (the simple eccentric diagram of Figs. 220 and 221), we emphasize the fact that the important result sought in any determination of valve motion is the movement of the valve edge with reference to the port edge, rather than of the center of valve with reference to center of seat — this by markedly substituting the lap line QR for the center line GH as a reference line. Further, we show in II how with positive lap the period of opening is less than half a revolution, so that the single valve is necessarily of this type, on the steam side at least; while the valve with negative lap, or the negative valve, opens its port for more than half a revolution, and can therefore control only one operation. Rotating II into the regular Reuleaux diagram position, we easily see that the eccentric driving a single valve must, of necessity, be in the second quadrant ahead of the crank if the valve is direct, in the fourth if it is indirect.

(*f*) Eccentric Setting for the Negative Valve. — In Fig. 255, Zeuner diagrams are used to illustrate these same ideas, and to show how the eccentric setting for a negative valve is determined. I and II, for the positive valve, are entirely obvious; but we now lay especial stress on the facts that the crescent EDG is a diagram of port opening, and that the cut-off intersection G determines the crank angle for which the valve is at the distance *s* to the right, but is moving toward the left.

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With the negative lap $-s$ shown at IV let us require that the valve close the port when the crank is at OC in III. The lap circle is drawn with absolute radius s, algebraic sign having no immediate significance. For cut-off, the valve must be to the left, wherefore the

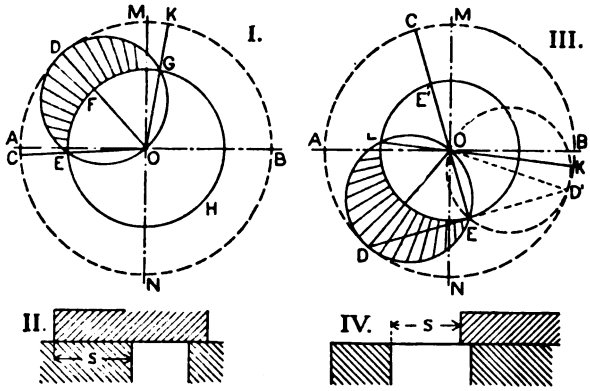

FIG. 255. — Locating the Valve Diagram.

valve circle must necessarily go through E instead of E′: and the fact that the negative travel intercept must increase as the crank advances puts the circle in the full-line position on OD, as against that on OD′. The crescent EDL is now a diagram of port closure; and we see further that it is characteristic of a negative cut-off valve, if of the direct type, to have its valve circle in the fourth quadrant, and its virtual eccentric in the third quadrant ahead of the crank.

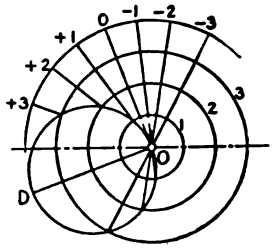

FIG. 256. — Control by Variable Lap.

(*g*) VARYING THE CUT-OFF. — The action of the device in Fig. 252 is shown by Fig. 256, where cut-offs are determined for laps varying by quarters from minus three-fourths to plus three-fourths of the radius of the virtual eccentric. Through comparison with Fig. 255 III, the diagram is self-explanatory. One point to be noted, especially marked for the cut-offs designated $+3$ and -3, is that the valve edge approaches the cut-off position more rapidly from the negative than from the positive side, the curvature of the valve circle showing this very clearly. Another point is, that with a large negative lap the period of closure is short; so that if the cut-off is very early, the expansion valve may reopen its port before the main valve cuts off. This

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possibility exists only with extreme proportions, and belongs rather to the method of regulation typified in Fig. 257.

In Fig. 257 I, with a negative lap equal to half of r_v, the valve circle is located for cut-off at 0.1, at 0.3, and at 0.5 of the stroke: and at II is given a diagram showing how the center E_2 of the expansion eccentric will have to be rotated about E_1 in order to secure this effect. It is a simple matter thus to pivot the second eccentric at the center of the first, and place it under the control of a shaft governor of suitable design. A wider range of cut-off would be covered than is here shown:

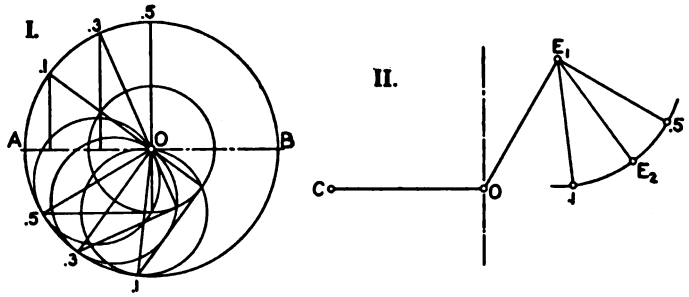

FIG. 257. — Rotating the Virtual Eccentric.

and it is apparent that the eccentric must be rotated by the governor through an angle equal to the crank angle between earliest and latest cut-off.

The scheme of Fig. 257 is the simplest case of varying the virtual eccentric, with the advantage that the second valve has a constant length of total travel upon the first valve. Another center can be chosen if desired, but the rotation of the cut-off eccentric about it will then cause a change in the length of the virtual eccentric.

PROBLEM

13. For the main valve, let $r_m = 2''$, $\delta_m = 120°$, $s_m = \frac{7}{8}''$, $i_m = \frac{1}{4}''$; for the cut-off valve, virtual eccentric $r_v = 1\frac{1}{2}''$, $\delta_v = 210°$. With these data make the following constructions:

A. Draw the diagram for the main valve, and show the entire steam distribution as determined by that valve.

B. Draw the virtual diagram for the expansion valve, and find the cut-off when $s_v = -\frac{1}{2}''$.

C. Drawing the two diagrams together, determine and show by sketches the positions of the two valves when the crank is at 0 deg., 45 deg., 90 deg., 180 deg., and 270 deg.: indicate by arrows the direction of movement of each valve, showing both the absolute and the relative motion of the riding valve.

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E. Make $s_v = +\frac{1}{2}''$, $0''$, $-1''$, and $-1\frac{1}{4}''$, and for each lap find the cut-off, as in Fig. 256.

F. Keep $r_v = 1\frac{1}{2}''$ and $s_v = -1''$, and change the eccentric angle so as to get cut-off at $\frac{1}{2}$, $\frac{5}{16}$, and $\frac{1}{8}$ of the stroke.

Draw separate figures for each requirement; use Zeuner diagrams.

(*h*) INDICATOR DIAGRAMS from engines with the double-valve gear are given in Fig. 258. The first set was taken from an engine with a load that fluctuated continually and widely, the effort to follow it keeping the governor dancing. The several expansion curves traced are selected from a large number of closely-spaced lines: constant compression and a quite sharp cut-off are the distinguishing features of these diagrams. At II the admission pressure drops as the piston speeds up toward mid-stroke, probably because the ports in this rather old engine are inadequate in size; but the cut-off curve is still quite

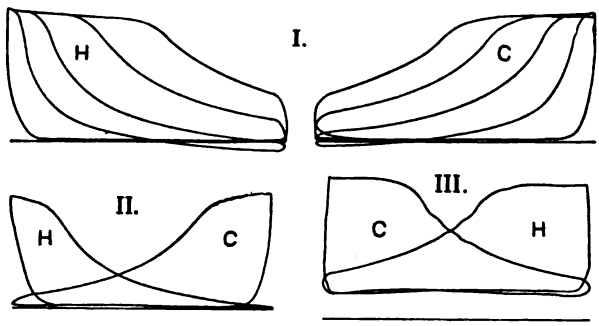

FIG. 258. — Indicator Diagrams from Engines with the Double-valve Gear. Engine I, 16 by 24 in. at 145 r.p.m.; II, 16 by 32 in. at 110 r.p.m., both in power service; III, air compressor, 10 by 12 in. at 145 r.p.m.

short. Rather different diagrams, with late admission and very short compression, are shown at III. Compare with all these the Corliss engine diagrams in Figs. 274 and 275.

§ 42. Details of Slide Valves and Gears

(*a*) CHARACTERISTICS OF VALVES. — The simplest possible plain slide valve is that used for illustrative purposes in Figs. 14 and 223: it makes provision only for the essential functions of the steam distribution. In practical application, this primary valve is modified and complicated along two lines: usually the valve is "balanced," or arranged so that it will not be forced hard against the seat by steam pressure; and very often the valve is so formed that it will admit steam past several edges, or is made "multiple-ported." The examples given

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here, together with those shown in some of the cylinder drawings at the beginning of § 36, are to be considered from the two points of view just indicated, namely,

First, as to form, with reference to the functions of steam distribution, and including the steam passages to the cylinder.

Second, as to balance and tightness, and incidentally the possibility of relieving excessive pressure in the cylinder.

(*b*) VARIOUS FORMS OF VALVE. — Figs. 200 to 202 and 259 to 263 illustrate the following descriptive classification:

1. The short, single-faced valve, with long cylinder ports, Figs. 200 201, and 262. This valve, mostly used in locomotives, has a balance rig on the back or top of it, working under a flat balance plate which is fastened to the steam-chest cover so as to be held parallel to the valve seat.

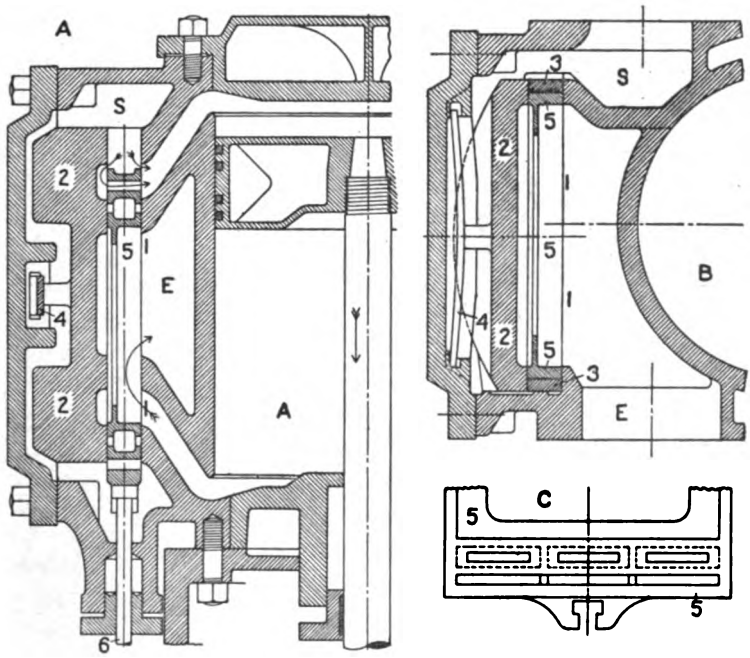

FIG. 259. — The Balanced Flat Valve, with parallel faces and double admission. A, Horizontal section; B, Cross section; C, Face of valve.

1. Valve seat.
2. Balance plate, very stiff against steam pressure.
5. Valve.
3. Distance strips, a very little thicker than the valve.
4. Spring, to hold plate in place when steam is off.

2. The long, double-faced valve, with short ports, Fig. 259. This works between valve seat and balance plate, with a very positive and

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very accurate mechanical fit. In long cylinders, and especially when the steam and exhaust ports are made separate, it is usual to divide the valve into two short slides, each under its own balance plate.

3. The double-seated valve, Fig. 202. In this arrangement there are really two complete valves and seats, the valves being set back to back and fitted together with a cylindrical, telescopic joint: the valve is indirect, having steam on the inside. The very long ports are a disadvantage of this scheme.

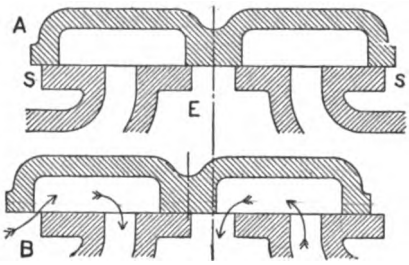

FIG. 260. — The B Valve.

4. The B valve, Fig. 260, is sometimes used in small steam pumps: the essential characteristic is that with the steam-chest arrangement of the direct valve, it has the movement of the indirect; this may be desirable in a gear of the type of Fig. 288.

5. The solid-plug piston valve, as in Fig. 9 and Fig. 263. In small engines (say up to 6 in. diameter of valve), the valve is often thus made

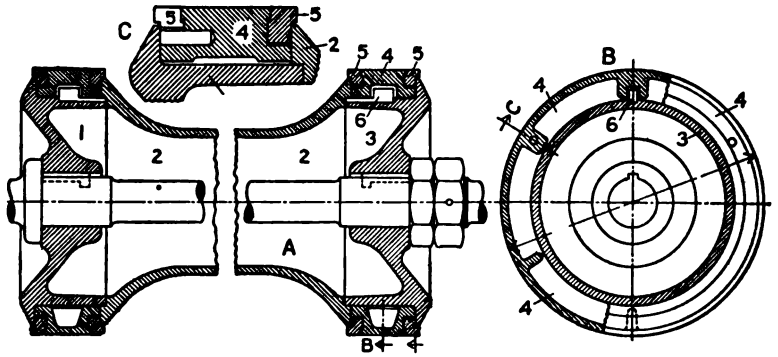

FIG. 261. — Piston Valve for Locomotive, with built-up style of construction: cylinder 20 by 28 in. Scale 1 to 8 and 1 to 4.

1, 3. Heads, held together by rod.
2. Valve body.
4. Bull ring or face ring.
5. Packing rings.

solid and finished to a close running fit: under favorable conditions (no grit and regular lubrication) the valve may run for a long time without enough wear to cause serious leakage. Sometimes adjusting devices are added, so that the valve can be expanded or the seat bushing contracted, to take up wear.

6. The packed or self-adjusting piston valve. In Fig. 261, there is at each end a solid bull ring (as on an engine piston like Fig. 206) and

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two overhanging spring rings — note how these are restrained by a narrow ledge, so that in case of breakage the pieces cannot get out into the ports. Besides rings which thus form the valve edges, common square rings, in grooves set a little way back from the edges, are often used.

(*c*) MULTIPLE-ADMISSION ARRANGEMENTS. — In Fig. 202 there is, as already remarked, simply a combination of two valves into one. The valve in Fig. 259 readily gives double admission, since it naturally uncovers two edges, and only a small port through the valve is needed to make the outer edge effective. Figure 262 shows how an equivalent

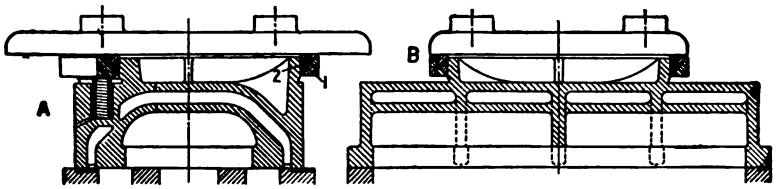

FIG. 262. — German Locomotive Valve with Allen Port. Scale 1 to 15.

result is got with a one-faced valve, but now a long extra passage, formed in the body of the valve, is required. Let the direct valve of Fig. 262 move to the right, or the indirect valve of Fig. 263 to the left: at the same time that the regular edge uncovers the port, the edge of the inner passage (at the other end of the valve) runs off the valve seat, so as to admit steam into this passage.

In general, the use of multiple-admission arrangements is much less common with the piston valve than with the flat valve. One of the purposes of the piston valve is to get long admission edges, with moderate overall dimensions and without complexity of arrangement.

(*d*) BALANCING DEVICES. — The flat valve in Fig. 259 and the solid piston valve in Fig. 263 are examples of what may be called rigid balance, dependent for tightness upon original accuracy of form and fit, and upon the prevention of wear. They are incapable of rising from the seat to relieve excess pressure in the cylinder: for while the valve in Fig. 259 is not mechanically constrained, the area of balance plate subjected to steam pressure is so great that a very high pressure in the port would be needed to push the valve away from its seat.

A flexible balance, as in Figs. 200 and 262, is less delicate, in construction and in service, than the arrangements just noticed, and is therefore far more suitable for the exacting conditions of transportation service, whether locomotive or marine. In Fig. 201, the fence or wall around the relieved area is made up of four strips which fit tightly against

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each other at the corners; note that the enclosed space is vented to the exhaust port. In Fig. 262 the relieved area is circular, as also in Fig. 202. Since the tightness of these valves depends upon their being pressed down on the seat by steam pressure, the area of relief is generally made something like 60 per cent of the whole area of the valve face, leaving the other 40 per cent more or less subject to the pressure of the

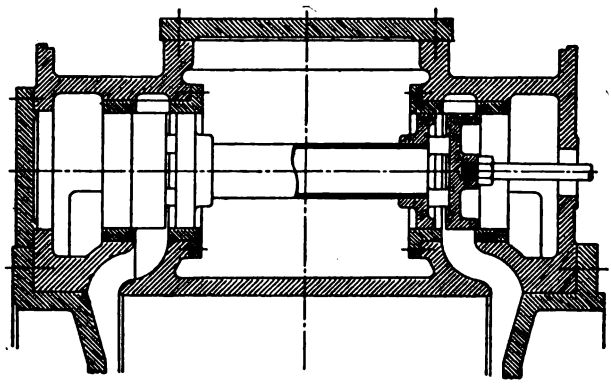

FIG. 263. — Piston Valve of the Allen Type.

steam. When a port is partly open, for admission, nearly or quite the full pressure acts upon the under face of the overlapping part of the valve, thus diminishing the effective unbalanced area.

(*e*) DESIGN OF STEAM PASSAGES. — The determining factor for the cross area of the port is the permissible velocity of the steam current. As the piston advances, the product of area A by instantaneous velocity v is the rate of volume generation by the piston; and in the steam passage, area a by steam velocity V is the rate of flow to fill the new volume. In terms of known quantities, then, the velocity V is

$$V = \frac{A}{a} v. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (207)$$

It is customary to proportion the ports for a maximum steam velocity of 160 to 240 ft. per sec., this maximum occurring near mid-stroke, where the piston is moving a trifle faster than the crank pin. Using 1.6 as the ratio of maximum to mean velocity of the piston — compare the statement preceding Eq. (150), page 293 — the range from 600 to 900 ft. per minute of mean piston speed corresponds with from 16 to 24 ft. per second of maximum velocity. In practice, the ratio of port area a to piston area A varies from one-tenth to one-eighth, in engines to run at from 500 to 800 ft. per minute of piston speed.

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In engines where the prevailing cut-off is early, the maximum speed of flow is seldom or never reached by the entering steam; but it always occurs during exhaust, hence this operation has the stronger determining influence; and if the ports are separated, as in Figs. 3 and 265, the exhaust port is made larger than the steam port. With a high terminal pressure, at the end of expansion, a great part of the exhaust steam will escape in the first release, while the piston is moving very slowly, and thereafter the velocity of the steam is definitely related to that of the piston.

Referring to § 16 (*e*), we see that the pressure drop required to produce the above-named velocities, through an ideal orifice, would be very small — far less than the loss between the steam chest and cylinder usually observed in engines. But other resistances have a greater effect than does the inertia of the steam: and this range of steam velocity has been found by experience to embody a good compromise between excessive pressure drop on one hand and excessive clearance volume on the other.

(*f*) PROPORTIONING THE VALVE. — The width of port having been fixed, the laps must be determined: this involves the whole question of steam distribution, and would be worked out by the methods of the preceding sections. With dimensions decided upon for the single face that covers one port, the combination into a complete valve is a simple matter, unless there is likely to be interference of function. Referring to Fig. 200, it is evident that if the valve is moved very far to the right, the inner or exhaust edge of the left face will begin to throttle the opening into the main exhaust port; so that there is a limit to the shortening of the valve. The B valve, Fig. 260, is subject to a similar and stronger limitation. In some of the more complex slide valves, not here illustrated, where there are several openings into one port, this question of preventing interference becomes a chief determinant of proportions.

(*g*) ECCENTRIC AND STRAP. — One example of these parts has been outlined in Fig. 242, while fuller detail is given in Fig. 264. The larger side of the eccentric disc or sheave is very much in the form of a wheel, with one or more arms. For large engines, the sheave is almost always in two parts, joined by bolts with nuts or keys. Cast iron is the usual material in stationary engines, with the smaller "half" sometimes of wrought metal; for locomotive and marine engines, cast steel is much used. In Fig. 242 the eccentric is secured by a key and set screws, while in Fig. 264 set screws alone are considered sufficient, and give an opportunity to change the setting of the eccentric.

The bearing between eccentric and strap, if not carefully fitted and

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kept well lubricated, may be one of the most troublesome bearings in an engine. The bearing surface is nearly always cylindrical (an exception being shown at H in Fig. 264), with recesses at the edges to make room for collars on the strap. In cruder lines of construction, there is a bearing of cast iron on cast iron, as indicated at E in Fig. 264; but it is better to line the strap with babbitt metal or brass, and with steel in sheave or strap this becomes necessary. Brass liners are shown at H and J in Fig. 264.

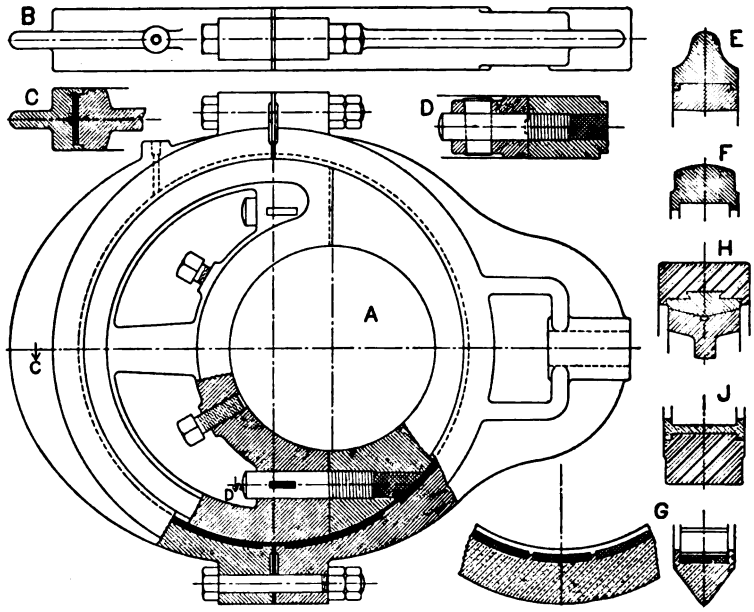

FIG. 264. — Eccentric and Strap for Corliss Engine in Fig. 3, to fit shaft in Fig. 215. Scale 1 to 14. Sketches E to J, various sections of the eccentric strap.

In that figure, the round eccentric rod passes through a yoke which projects from the strap, and is fastened by two nuts, one in the recess within the yoke. Very often a T end is formed on the rod and it is bolted to the strap, in an arrangement closely analogous to the marine type of connecting-rod end shown in Fig. 2. The locomotive, as in Fig. 242, has stiff rods, of deep rectangular cross section, very strongly fastened into a nose on the strap: there is much possibility of failure of lubrication, and the rod must be strong enough to keep the strap turning on the sheave, even against a high resistance.

(*h*) RODS, ROCKERS, ETC. — In some valve gears, notably on locomotives, the rods are made with solid forged ends, the eyes or bearings being lined with hardened steel bushings and the pins case-hardened.

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On most stationary engines, adjustable rod ends are used, similar to those of connecting rods, with brass bearings. As shown in Fig. 269, at the rod 5, the end or head is most frequently a separate piece, into which the rod body screws; by using right and left threads, with lock nuts, and providing a hold for a wrench, adjustment of length is made easy.

Occasionally a small slide block is used at the inner end of the eccentric rod, corresponding with the crosshead; but the rocker arm, of which examples have been quite fully shown in Figs. 5 and 6, is much more common. As to the function of the rocker, it may simply guide a joint as in Fig. 12, may change amplitude of motion as in Figs. 3 and 267, or may reverse motion as in Figs. 229 and 244: further, it may be used to transfer motion sidewise from one plane (perpendicular to the shaft axis) to another. With such transfer, the direct rocker takes the form of the letter U, the indirect of Z.

§ 43. The Corliss Valve Gear

(*a*) THE CORLISS CYLINDER. — A representative example, with valves of the simplest form, is shown in Figs. 265 and 266; in the end view some of the valve-gear parts appear. Note how the T head of the valve stem or spindle is engaged in a slot in the end of the valve, and how the yoke extending from the valve bonnet forms a support and bearing for this spindle. The wrist plate is carried on a heavy stud which is bolted to a seating on the cylinder.

(*b*) THE VALVE GEAR of this engine is shown in Fig. 267, partly in skeleton: the mechanism on the cylinder is made most prominent, but the shaft outline (the crank-eccentric COE) and the rocker arm are drawn in true proportions at A and B. By means of eccentric rod 1, rocker arm 2, and reach rod or hook rod 3, motion almost harmonic is given to the point H on the wrist plate 4; and we may say that this piece oscillates in practically harmonic motion, conceiving its angular movement as determined by the linear movement of H. The reach rod is not permanently connected to the wrist plate, but merely hooks over the pin, so that the valves can be moved by hand, with " starting bar " S, in starting and stopping the engine.

Through the valve rods 5 and 7 and the cranks 6 and 8 the valves are given an oscillation which is far from harmonic. Intentionally, the opposite angular displacements of any valve arm, from a mid-position determined by putting the eccentric at 90 deg., are made very unequal. This is clearly shown in Fig. 270, where the mechanism is drawn in this mid-position and the range of movement of each C-point is marked.

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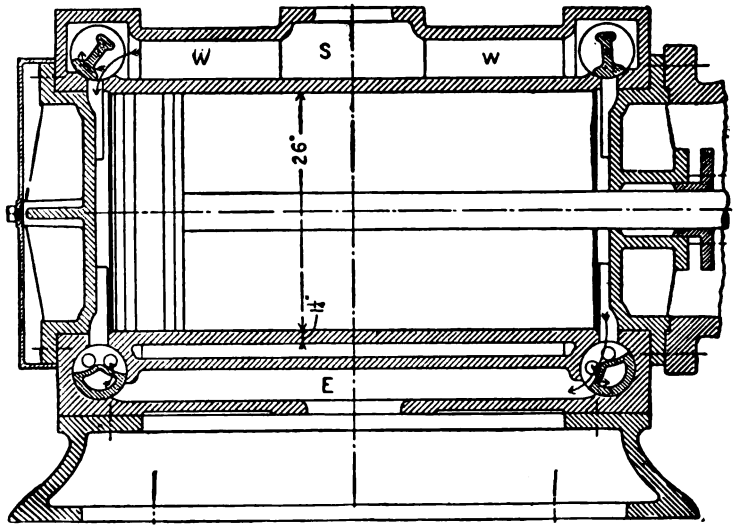

FIG. 265. — Lengthwise Section of Corliss Cylinder, 26 by 48 in. Scale 1 to 21.

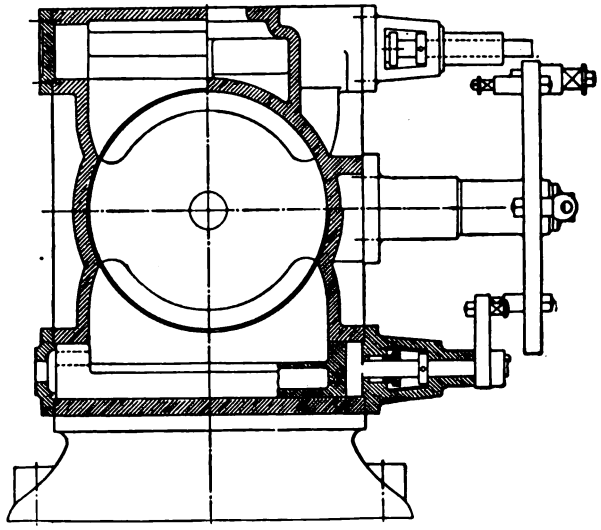

FIG. 266. — Cross Section of the Cylinder in Fig. 265. Upper right-hand quarter taken at mid-length, the rest of the section through the head-end ports.

The result sought and obtained is, that the valve shall have a wider and quicker movement in the direction for opening, a shorter and slower movement on the closure side.

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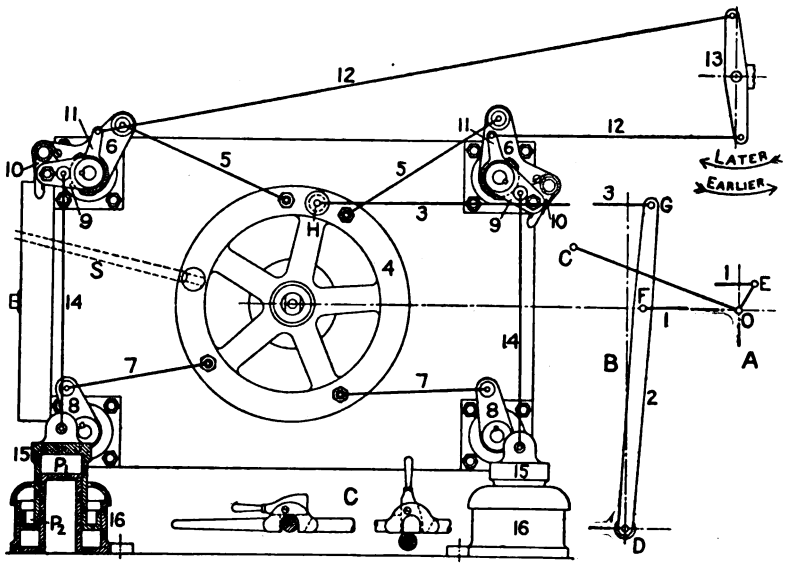

FIG. 267. — Corliss Valve Gear, Single-eccentric Type, with Full Wrist Plate.

1. Eccentric rod.
2. Rocker arm.
3. Reach rod.
4. Wrist plate.
5. Steam-valve rods.
6. Oscillating cranks.
7. Exhaust-valve rods.
8. Exhaust-valve arms.
9. Steam-valve arms.
10. Hook claw.
11. Cam ring.
12. Governor rods.
13. Governor rocker.
14. Dashpot rod.
15. Dashpot plunger.
16. Dashpot body.

(c) THE CUT-OFF MECHANISM. — The releasing gear for the head-end steam valve is shown in detail by Fig. 269. Valve arm 9 is merely dotted in, although the hook pin P, carried by it, is drawn in full. At B the several cranks are shown as if swung into one vertical plane, which involves a distortion of 6; and to save overlapping the hook claw or latch 10 is represented only by its center line. All the valve rods have heads like that on 5, and are adjustable in length by means of right and left threads, as are also the governor rods 12 and the drop rods 14. The relative positions and the different motion planes of the parts of the whole mechanism have already been illustrated in Fig. 266.

The oscillating crank 6 has its bearing on the valve bracket or bonnet B. While it is turning toward the left, the valve is closed from the previous cut-off, and at rest; as 6 nears the limit of travel, the latch L slides over the catch pin P, going just a little way past the engaging point. As the crank is pulled toward the right it turns the valve stem with it, until the trip arm T strikes the knock-off cam C_1 and L is pushed out so as to disengage P; whereupon the dashpot pulls the valve quickly back to its rest position, giving a sharp cut-off.

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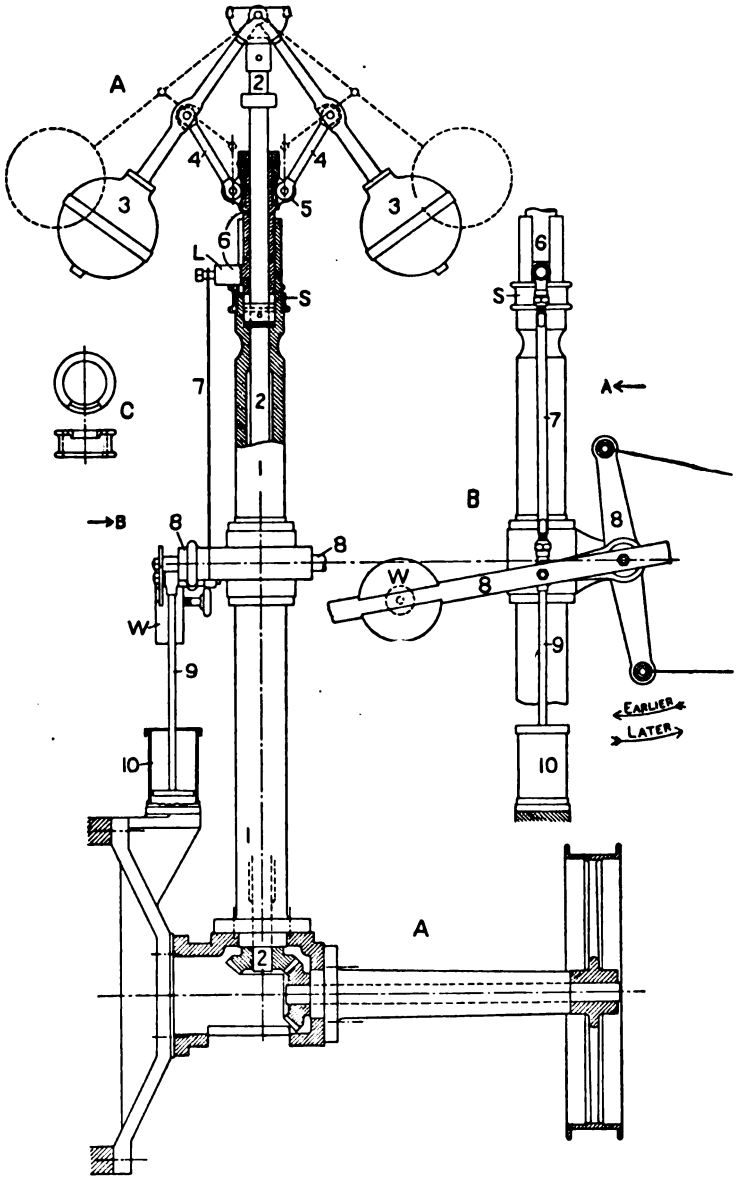

FIG. 268. — The Governor.

1. Governor standard.
2. Governor spindle.
3. Weight arms.
4. Lifting links.
5. Lifting ring.
6. Lifting sleeve.
7. Lifting rod.
8. Governor rocker.
9. Dashpot rod.
10. Dashpot.

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It will be noted that all the hooking and tripping surfaces are carried by small pieces of hardened steel, shaded on the figure, which can easily be readjusted or replaced — being held fast by small bolts or screws not fully shown on the drawing. A section of one dashpot is given in Fig. 267; it is of the double type, the inner plunger P_1 being mostly concerned with the vacuum action, the annular plunger P_2 with cushioning. A fuller description and discussion of the dashpot will be found in Art. (*j*).

(*d*) CONTROL OF THE CUT-OFF. — The manner in which the time of cut-off is determined by the governor, through movement of the cam ring 11, is apparent from Figs. 267 and 269. An important point is that the trip arm must strike the cam C_1 before the crank 6 gets to its

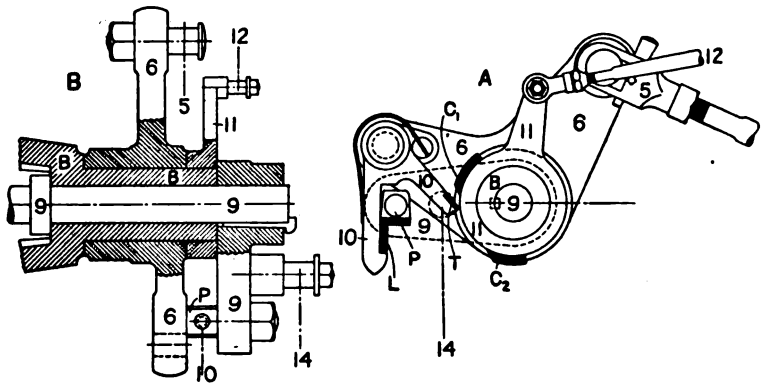

FIG. 269. — Detail of the Releasing Gear.

limit of travel on the open side; otherwise the valve will not be released at all, and steam will be admitted through nearly the whole stroke. This means that the latest cut-off operation under control of the governor must begin a little while before the eccentric gets to its dead point — rapidity of closure then depending upon the strength of the dashpot. As will presently be shown, this places the limit of controlled cut-off at from 40 to 45 per cent of the stroke in engines which have all their valves operated by one eccentric.

The Governor. — This is of the vertical fly-ball type for a Corliss engine, the one from the machine under consideration being shown in Fig. 268, where view A is taken from the cylinder, looking toward the shaft, while B is a view from the front of the engine, or the side opposite the valve gear. The working of the governor mechanism is self-evident, the sleeve 6 being moved up and down as the centrifugal force of the balls varies with reference to the downward pull of their own weight and of the balance weight W; and the running speed can be changed

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by moving W in and out along its arm. The dashpot 10, with a loose-fitting piston working in oil, is of the drag or damping type, and is put on to keep the governor from responding too freely to small and irregular impulses. The governor rocker 8 is the piece, numbered 13 on Fig. 267.

Action of the Safety Cams. — In Fig. 268 the whole mechanism is drawn in the starting position (for the engine standing idle), with the arm L resting upon the stop ring S. This gives the latest possible cut-off, without release, the trip arm T, Fig. 269, just working between cams C_1 and C_2. The ring S, freely turned by hand after the governor

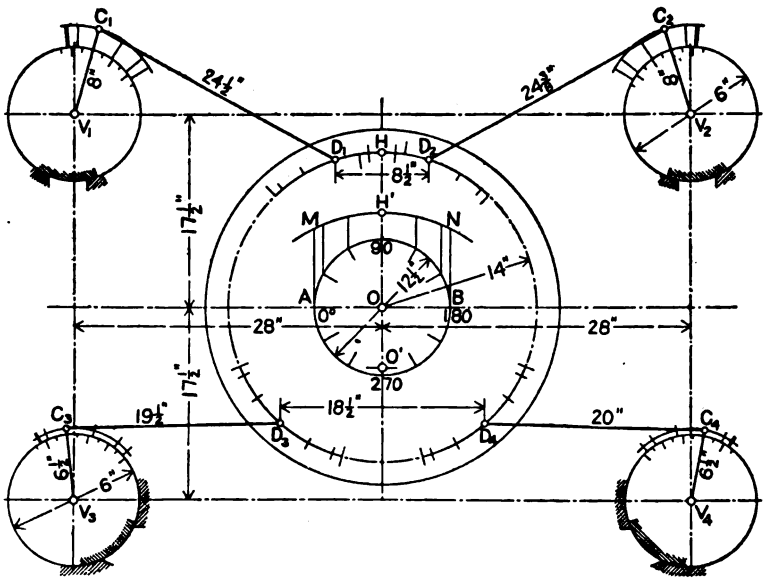

FIG. 270. — Diagram of the Valve Gear.

has lifted, has a part of its upper edge cut away, as shown at C on Fig. 268. When the engine is running, this notch is to be brought under L; then if through accident to its belt or for some other reason the governor ceases to turn, the balls will drop below the rest position, the safety cam C_2 will be brought around so far as to prevent engagement of the valve arm and opening of the valve, and the engine will be shut down. Further, when the governor is at its highest position the knock-off cam C_1 should release the valve before it is turned far enough to open the port, and thus prevent any admission of steam.

This simple safety stop has the drawback that if through overload or fall of boiler pressure the speed drops below normal, the arm L will

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sink into the notch in S, and the engine will be shut down. To prevent this, the engineer may not turn the ring into the safety position, thus depriving the engine of protection against the effect of an accident to the governor. More fully developed devices connect the support for L to an arm at the end of which is a small idle pulley riding on the governor belt; if the belt breaks, the stop is pulled aside, and the governor can drop to the low position and shut off steam.

(*e*) MOVEMENT OF THE VALVE. — In Fig. 270, the mechanism on the cylinder is drawn in mid-position, or with the eccentric at 90 deg., and a series of other positions is plotted, for the eccentric at equally-spaced angles. The results, in the form of ordinates which represent the linear travel of a point on the circular valve profile, relative to a fixed point on the seat, are laid out in Fig. 271, with the developed or unrolled eccentric circle as base. For the steam valves we at first disregard the releasing gear, either imagining the oscillating crank to be fast to the valve stem, or considering that we determine the movement of a point on this crank projected out from the valve face. The dimensions necessary for a layout of the valve gear are shown on Fig. 270: this particular example was measured up from the actual engine, and it will be noted that the valve rods are adjusted to different lengths.

The circle on AB is first drawn, with a radius equal to that of the eccentric when reduced to the upper end of the rocker arm, or when multiplied by the ratio GD/FD from Fig. 267. Swing of the eccentric rod may be disregarded, and the horizontal movement of the pin H be taken as harmonic. Then an angle scale for the movement of the wrist plate is got by dividing the eccentric circle and projecting the points of division to the path of H. For clearness of drawing, this path is moved down to MN, on an arc struck from O′ with a radius equal to OH.

The four driving points D_1, D_2, D_3, D_4, being all on the same circle with H, this travel scale MN is next centered on each one: where two paths overlap, as is the case with D_1 and D_2, one scale is marked outside, the other inside, of the circle. The positions of the C-points are now found by striking off the rod lengths, and are then projected radially upon the circles representing the profile of the full cross section of the valve.

Figure 270 is the picture of a drawing in which the mechanism as a whole was laid off half size, but the valves drawn full size: then the actual travel of the valve surface is given by the operation last described, and it is only necessary to rectify the curved path in order to have the desired ordinates of travel. For this purpose a scale of inches laid off on the valve seat, here marked on the inside of each circle, and

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used in connection with the horizontal ruling on Fig. 271, is most convenient.

The movement curves got by plotting the several sets of ordinates are given in Fig. 271, I and II for the steam valves, III and IV for the exhaust valves. The angle scale at the top of the diagram is the same as that on the circle AB in Fig. 270. Travel in the direction which opens the port is represented by an upward ordinate for the head-end curves I and III, by a downward ordinate for II and IV. The distortion from harmonic motion is very clearly shown by these curves.

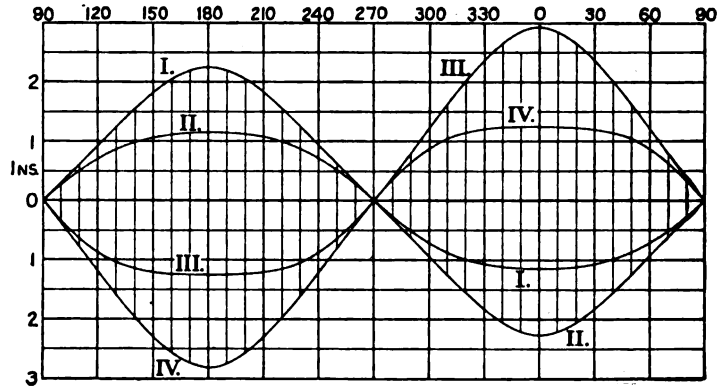

FIG. 271. — Valve-movement Curves.

(*f*) ECCENTRIC SETTING AND VALVE ACTION. — Having drawn the simple curves of valve displacement in terms of eccentric position, our next step is to combine with these the lap of the valves — measured in the usual way, in the position shown on Fig. 270 — and thus find out the proper setting of the eccentric with reference to the crank. For present purposes it is enough to consider one end only of the cylinder, wherefore curves I and III are reproduced in Fig. 272. It is the operation of exhaust that determines the eccentric setting, because both ends of this period have to be considered, as against only the beginning of the period of admission. Usually the lead for release, the angle TOB on Fig. 223, is less than the angle of compression, so that a small positive lap on the exhaust valve is necessary: this is represented by the distance of the line TS above the base line, being here one-quarter of an inch. Since the lead for admission must be less than that for release, the steam lap must be larger, and QR is drawn at three-eighths above the base line. We now locate suitably the dead-center positions of the crank, A for head end, B for crank end, and find the angle δ to be about

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106 deg. or the angle of advance about 16 deg. — this by noting that when the crank is at A or zero, the eccentric is at 106 deg. on its scale as marked below the base line.

The Cut-off Action. — Instead of following the movement represented by the whole of curve I, the steam valve really moves according to the full-line curves 1, 2, 3, or 4, of which the portion CDE shows

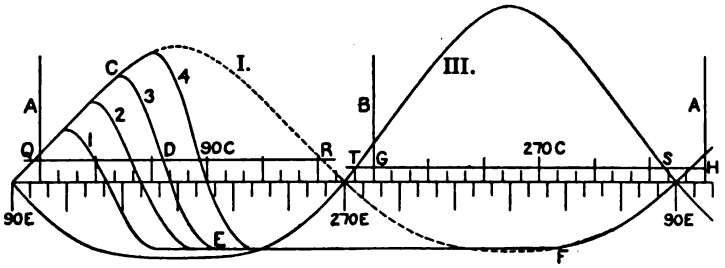

FIG. 272. — Diagrams for the Head-end Valves.

quick closure under the pull of the dashpot, and the straight line EF the period of rest while waiting for the next admission. These curves are merely sketched in from general considerations, since a mathematical determination of the action of the dashpot, while not incapable of giving quite accurate results, would be, in length and complication, rather out of proportion to the importance of the subject. Only a roughly approximate calculation would be made in working out a design, to adapt the results of experience to the particular case.

Valve and Indicator Diagrams. — The steam-valve curves from Fig. 272 are redrawn in Fig. 273, upon the stroke line of the piston as base, the

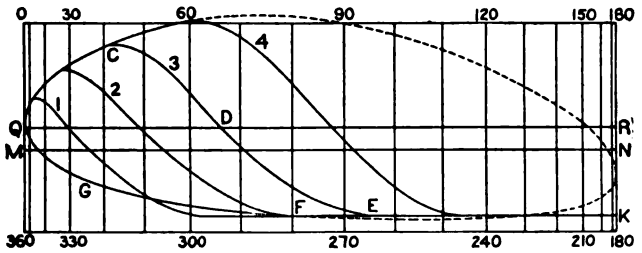

FIG. 273. — Stroke-line Diagrams for the Steam Valve.

piston positions being found as in Fig. 163 or Fig. 224. With these are to be compared the autographic diagrams in Fig. 274 I, which were taken from this same engine, being drawn by a pencil moved by the drop rod and working on the drum of an indicator which was con-

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nected to the ordinary reducing motion from the crosshead. Simultaneous indicator diagrams are given at II. For the head end, the two types of diagram agree very well in proportions, the short vertical lines showing where cut-off takes place. These are first located on II, then transferred to I with due regard to the difference in length of the diagram. The crank-end diagrams were evidently not really taken at the same time.

The characteristic Corliss-engine diagram is well represented by Fig. 274 II: the distinguishing features are, a horizontal admission line,

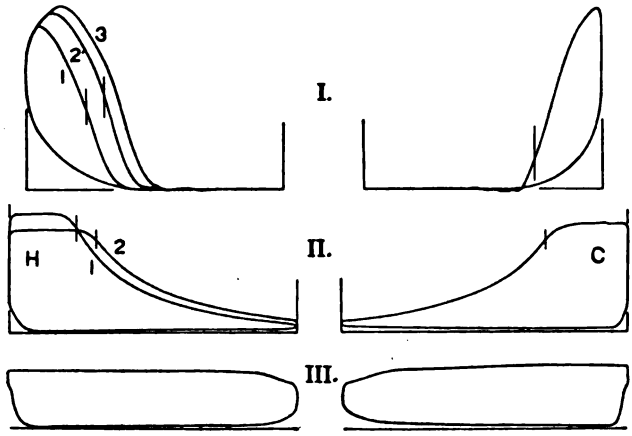

FIG. 274. — Indicator Diagrams from the Engine in Fig. 267.

a sharp and usually rather early cut-off, release deferred till near the end of the stroke, and a compression curve which begins late but rises rapidly on account of the small clearance. At III is shown what happens when the boiler pressure drops so low that the engine cannot keep up to speed under its load, and the releasing gear fails to act — the steam consumption per revolution being now much greater than when the plant is working properly.

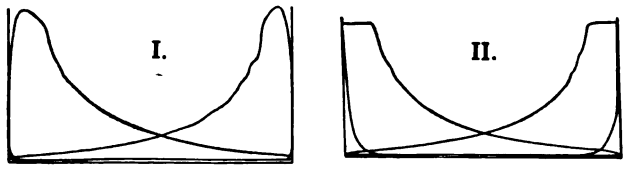

FIG. 275. — Faults in the Eccentric Setting.

A great deal of the valve setting to be done on a Corliss engine consists simply in adjusting the lengths of the various rods. Fig. 275, however, shows faults in the setting of the eccentric: at I it needs

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to be advanced, at II we see the effect of advancing it too far, apparent especially in the excessive lead shown by the outward slant of the rising admission lines.

(*g*) VALVE RESISTANCE IN THE CORLISS GEAR. — The amount of work that must be expended in moving a set of Corliss valves is small, not so much because the valves are not at times pressed very hard upon their seats as because the movements under this heavy pressure are small. The steam valve does resist strongly while being opened; but it need not be closed by any greater amount of overlap than required to insure tightness — Fig. 272 showing more closure travel than is really necessary. When it is once open, any single-function valve is balanced or left free to move easily by the substantial equalization of pressure all around it. For the exhaust valve especially there is a decided gain in a marked distortion from harmonic motion. Referring to Fig. 272 we see that the period of rest and of slow movement of this valve coincides with the time when the steam pressure in the cylinder is high; and the wide and rapid movement takes place after the pressure has been lowered by expansion, and while the port is open.

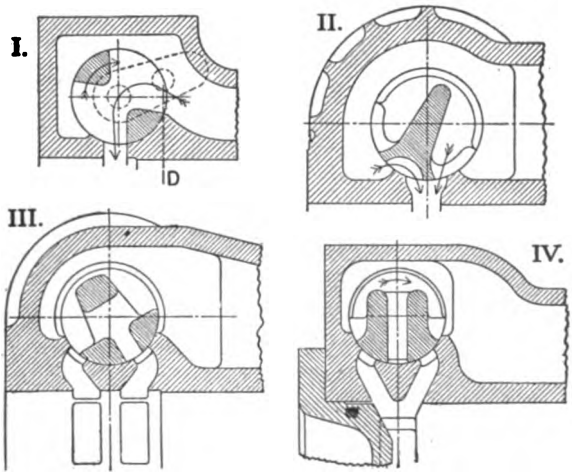

FIG. 276. — Various Steam Valves: I. Passage through Valve; II. Murray Valve; III. Brown Valve; IV. Reynolds Valve.

(*h*) VARIOUS FORMS OF THE CORLISS VALVE. — The sections in Figs. 276 and 277, with Fig. 265, illustrate the following points:

1. If the steam valve opens by an inward motion across the port (like the direct slide valve), as in Fig. 276 I and III, the drop rods being on the inside of the valve stems, the engine has outside admission; the opposite arrangement, in Figs. 265 and 267, and at II and IV of Fig.

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276, gives inside admission. With outside opening, the steam has a little longer path to traverse; and the purpose of the passage through valves I and III of Fig. 276 is to make this path as short and direct as possible.

2. In Fig. 276 II and Fig. 277 III (both from Fig. 3), double opening is secured with a single port, by using the principle of the B valve, Fig. 260, for the second opening.

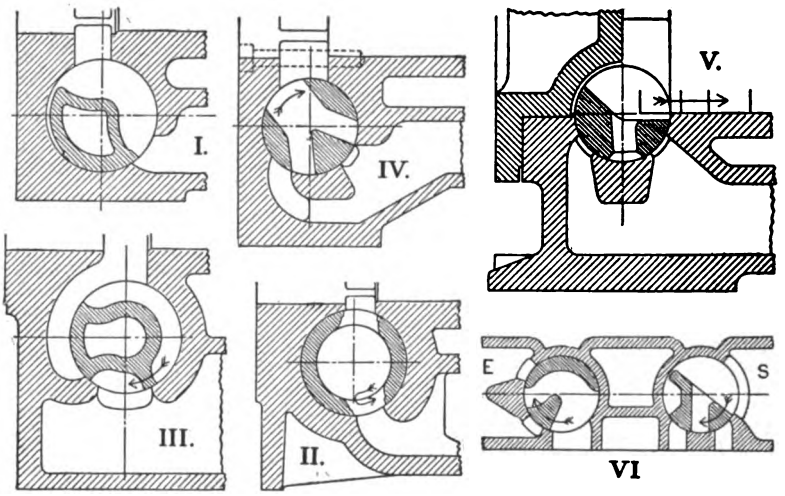

FIG. 277. — Various Exhaust Valves: I. Plain Single-ported Valve; II. Fleming Valve; III. Murray Valve; IV. Reynolds Valve; V. Brown Valve; VI. Valves in Head of Cylinder.

3. Double-ported valves are given in Fig. 276 III and IV and in Fig. 277 IV, V, and VI. Besides benefiting the steam distribution, the purpose of all these designs is to decrease the movement of the valve gear, thus adapting the engine to easy operation at higher speeds. The usual limit for Corliss engines is from 110 to 120 r.p.m., although the gear can be operated definitely, if not quietly, at as much as 150 r.p.m.

4. The exhaust valve is contained within the engine cylinder, so far as steam space or clearance volume is concerned. Valves I and II in Fig. 277 are made hollow and bulky, so that they fill all of the space, within the cylinder of the valve, that is not needed for the passage of steam; while valve No. V is moved up into the cylinder space, where it just nicely clears the piston when fully closed. No. II shows an attempt to make the valve close the port which enters from the cylinder, instead of, or in addition to, that which opens into the exhaust chamber;

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the probability of success in this effort, depending upon the maintenance of a tight fit, is low.

5. In large engines, especially when vertical, the valves are commonly placed in the heads, as in Fig. 277 VI; this makes the ports short, and greatly simplifies the cylinder casting.

(*i*) THE USE OF TWO ECCENTRICS. — As already stated in Art. (*d*), the valve must be released before the eccentric gets to its dead point; and in Art. (*f*) it is shown that an eccentric to drive exhaust valves must have some advance, usually 15 to 20 deg. This condition is represented in Fig. 278 I, where δ has the minimum value 105 deg., and it is assumed that latest dependable release must take place 15 deg. before dead center of the eccentric, or when the crank is at 60 deg.: this corresponds very closely with curve 4 on Fig. 272.

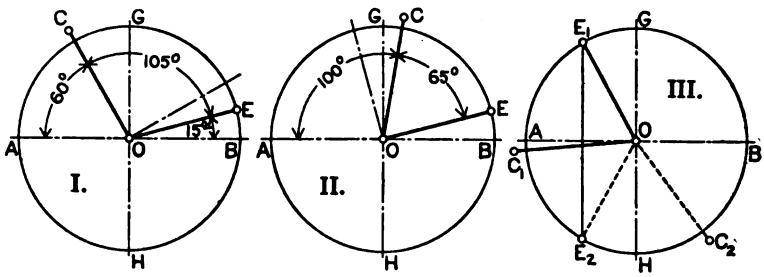

FIG. 278. — Diagrams of Eccentric Setting.

A very obvious way to increase the range of cut-off is to set the eccentric back, toward the crank, so that it will turn through a larger angle from the position for crank on dead center to that for latest release. In Fig. 278 II the negative advance or the angle of lag is 25 deg., and the latest cut-off begins with the crank about at 100 deg. It is entirely evident that this eccentric can be used for the steam valves only.

The limit to the setting back of the eccentric is found in the fact that if the negative lap is made too great, the valve will not cover the port effectively when closed. Thus in Fig. 278 III, C_1OE_1 is the crank eccentric from II in the position for beginning of admission, and we see through what a small distance the valve has moved from its rest position — a distance which is all the smaller if there has been much distortion from harmonic motion on the closure side. The last diagram brings out further the absolute necessity of insuring that a gear of this type shall never fail to release: for the condition which produced the indicator diagrams in Fig. 274 III would in this case keep the valve open until the crank got around to C_2, or far into the exhaust period.

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A fairly obvious scheme for getting a longer range of controlled cut-off with one eccentric is to give the trip cams a cyclical motion, instead of simply having them held in one position by the governor. Then even after the valve arm has begun to return from its extreme position, the moving cam may overtake it and release the valve. This idea is applied in at least one design of power engine, and in several pumping engines — see STEAM ENGINE, Vol. II, pages 320 to 324.

(*j*) THE DASHPOT. — The primary function of this device is to exert a sharp and strong force, which will close the valve quickly; the name comes rather from the secondary function of cushioning the

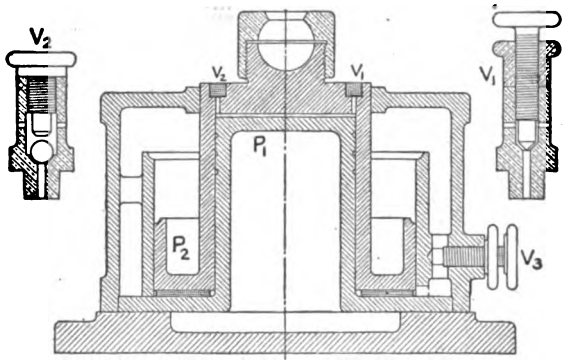

FIG. 279. — A Two-plunger Dashpot.

movement, or bringing the parts quietly to rest. A dashpot essentially the same as that in Fig. 267 is shown in Fig. 279: sometimes a single plunger is used, but with the double arrangement there seems to be more ease and flexibility of control.

As the plunger rises, a partial vacuum is formed beneath it, and some air is drawn in; as it nears the bottom of the cylinder, on the drop, this air is compressed and furnishes the needed counterforce. On the inner cylinder there is first the adjustable valve V_1, next the check valve V_2; so that the egress is made freer than the ingress, because the downward movement is quicker than the rise. The inner plunger is intended to exert the stronger vacuum action, the outer is more concerned with cushioning. Relative to the piston displacement which it serves, the valve V_3 is larger than the other two together, so as to permit readier inflow of air and cause a stronger checking of the descent. Both as to form and proportions, this device is based very largely upon experience and trial.

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§ 44. Various Valve Gears

(*a*) RELEASING GEARS WITH GRIDIRON VALVES. — There are a number of standard engines having a steam distribution exactly equivalent to that of the Corliss gear, but using flat gridiron valves. In location of valves (on top, at side, or at bottom of cylinder), and in form and detail of mechanism, these show a wider variation than do the more numerous designs which have been developed from the original Corliss. Sometimes the valves are short and wide, serving only two or three openings into the port, oftener they are narrow and long (length being taken in the direction of movement), covering a large number of short openings which lie across the port. The piece which moves the valves may be a reciprocating slide driven by an eccentric in the usual fashion, a shaft parallel to the engine axis and receiving an oscillating motion from an eccentric whose rod drops vertically to an arm on this shaft, or a rotating lay shaft, as in Fig. 282, which carries individual eccentrics at the valves.

(*b*) NON-HARMONIC GEARS WITH SHIFTING ECCENTRIC. — To remove the limitation of speed to which the releasing gear is subject, while retaining the advantage of non-harmonic motion with separate, single-function valves, engines are built with the cylinder and valve arrangement of the Corliss and allied types, but with the steam valves driven by a shaft governor while a fixed eccentric drives the exhaust valves. A steam valve thus controlling both admission and cut-off must give narrow port opening with early cut-off, just as does the slide valve. The extreme of development along this line is seen in the McIntosh & Seymour gear, which has a double-valve arrangement on the steam ports; the riding valve is driven from the shaft governor, and all the valves are of the second gridiron form defined above. The engines represented by Fig. 97 are all of this design For analysis of these gears, see STEAM ENGINE, Vol. II, pages 332 to 347.

(*c*) THE LIFT-VALVE ENGINE. — For general power service, engines with lift or poppet valves have the same predominant position on the Continent of Europe that the Corliss engine has in American practice. The commonest arrangement of the valves, shown in Fig. 203, is similar to that of the Corliss cylinder in Fig. 265. Sometimes the valves are placed at the sides of a horizontal cylinder, or in the heads: and one or the other of these locations is used in vertical engines, the valve axis being always kept vertical. A very full description of the form and working of this gear, in all its variations, will be found in *Steuerungen für Dampfmaschinen*, by Leist. Only a few prominent points will be noted here.

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(*d*) TYPES OF VALVE. — Poppet valves are usually double-seated, and are so arranged as to be nearly balanced, only an annulus equal to or slightly exceeding the combined width of the contact surfaces being subjected to downward pressure when the valve is closed. The solid double-disc valve at I in Fig. 280 is the simplest in its own form, but

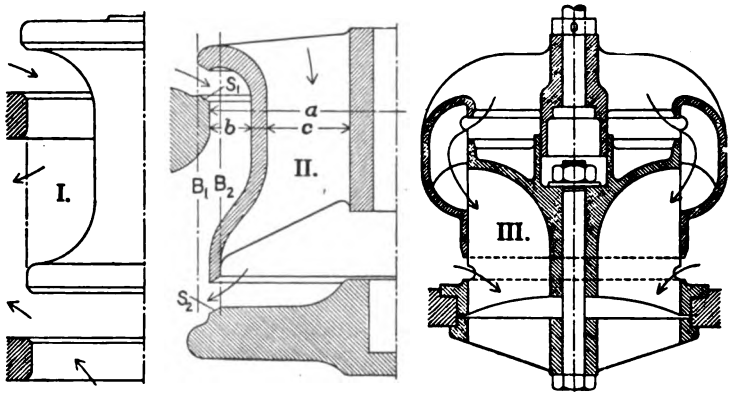

FIG. 280. — Different Types of the Two-seated Valve. I. Valve with double inlet; II. Valve with single inlet; III. Bell or Cornish valve.

requires a more complicated valve chamber, because the steam coming to it must be given access to both top and bottom, by means of passages formed in the casting; but for a given diameter it has a larger capacity than the valve which receives steam from the top only, as at II. The latter, taken from Fig. 203, has the determining dimensions marked upon it: the effective opening is the area of the circle with diameter *a*, less the cross-sectional area of the valve; and the annular passages of the widths *b* and *c* should be given equal areas, so as to permit equal flow to or from both openings. Type III is derived from II by a kinematic inversion, the valve in one case corresponding in essential form with the seat in the other: it has the advantage that fitting the valve seat into the cylinder is a much simpler matter than with the double-contact arrangement in Figs. 203, 283, and 284; but it appears to require rather more room, so that the volume of the cylinder clearance will be greater.

These valves as well as the inserted valve seats are made of hard cast iron. The contact strips or seats are narrow, ranging with the size of the valve, from one-eighth to one-half inch in width; they vary from a 45-degree cone as in I to a plane surface as in III. Almost always

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the valves are arranged to lift in opening, although engines have been built in which some of the valves open downward and are held up by springs.

Sometimes single-disc poppet valves are used in the low-pressure cylinders of pumping engines. Thus the triple engine whose diagram is given in Fig. 131 I has these valves for exhaust on the low-pressure cylinder; and the quadruple of Fig. 140, with eight sets of valves in all, has single-disc valves for the last three. For some large engines, four-seated poppet valves have been used successfully.

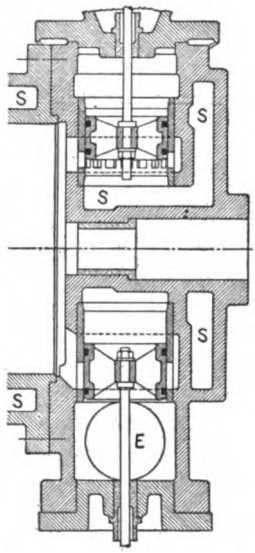

FIG. 281. — Van den Kerchove Piston-lift Valves.

A comparatively recent variation from the usual lift-valve practice (introduced about the year 1900) is shown in Fig. 281. These piston valves do not have to be brought to rest exactly at a certain level, and therefore require a less precise adjustment of the valve gear, if the latter cuts off by release: they open at only one edge, but with the greater facility offered for absorbing their momentum quietly, they can be lifted higher and dropped more sharply than the double-opening poppet valves.

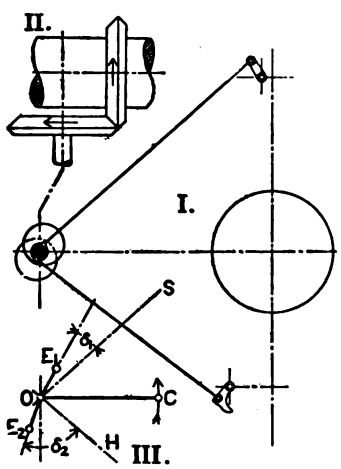

FIG. 282. — Outline of Gear at Cylinder; General Arrangement as in Fig. 203.

(e) GENERAL ARRANGEMENT OF THE VALVE GEAR. — With the valves located as in Fig. 203, the gear at the cylinder takes the form outlined in Fig. 282. In I, the large circle represents the bore of the cylinder, and the lay shaft is shown in section, with the two eccentrics at one end of the cylinder; II shows how the shaft is driven by miter gears, and the eccentric setting is laid out at III. Here the crank is shown as if on the valve-gear shaft and at dead center, and each δ as marked is measured from the stroke line (the mean direction of the eccentric rod) to the radius arm OE. The steam eccentric OE_1 is only far enough ahead of the crank to take up the over travel of the latch in Fig. 283, and give the valve a small lead when the crank is on dead center:

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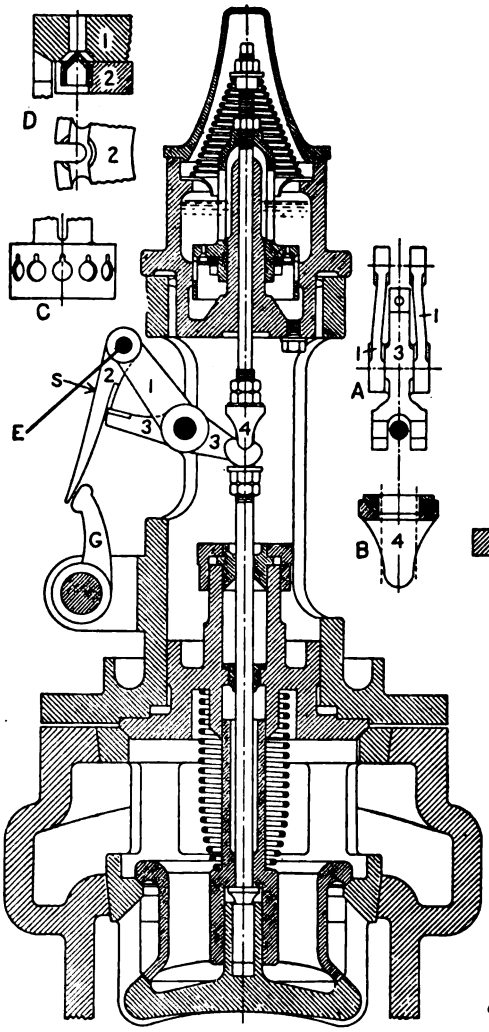

FIG. 283. — The Collman Releasing Gear, High-pressure Cylinder of Compound Engine. Scale 1 to 8.

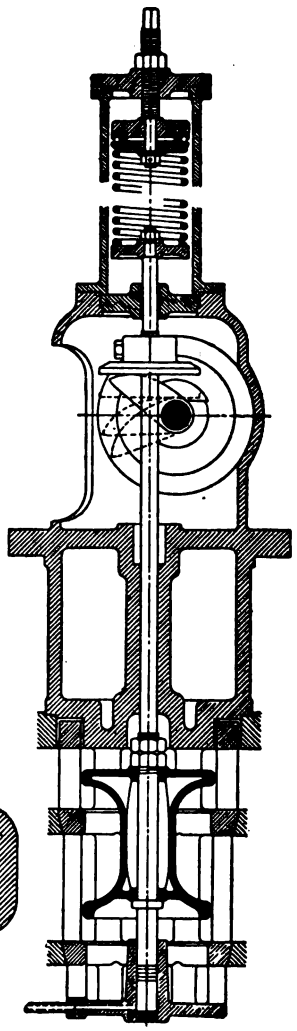

FIG. 284. — Steam Valve and Gear, 42 by 60 in. high-pressure cylinder of engines in Subway Power House, New York. Scale 1 to 18.

the exhaust eccentric OE_2 has about the same angle of advance (from a perpendicular to OH) as for a Corliss engine — somewhere near twenty degrees.

(*f*) ADMISSION VALVES WITH RELEASING GEAR. — A first-rate German design is shown in Fig. 283. With the help of the detail at A, the arrangement of the driving mechanism is clear; at B we see how

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piece 4 is made free to oscillate with the swing of 3. The forces which act to close the valve are the weight of all the parts attached to the valve stem and the push of the lower spring, here placed in the steam space. A rapid diminution of this force as the valve nears its seat is secured by the use of the lighter counter spring at the top; and the movement is checked by an oil dashpot, which has its plunger formed as better shown at C. The large holes let the oil pass freely while the valve is up, but only a small opening is effective when the valve is close to the seat. To avoid having the same oil resistance at the beginning of the rise as at the end of the drop, a number of small check valves are placed in a circle in the disc of the plunger, being made somewhat as suggested by the sketch at D. Air dashpots are sometimes used, and it will be noted that the lower end of the valve stem and guide on which the valve hub slides will act as a small dashpot with steam. Of course, to bring the drop valve quietly and yet quickly to its seat is a more delicate task than the closing of the Corliss or other sliding cut-off valve.

In Fig. 284 the valve is raised by a rocking cam, carried on a spindle which is oscillated and released by the usual Corliss gear, with the dashpot on the side of the cylinder. The cam is forged solid with the spindle, and is forked so as to bear under the collar on both sides of the valve stem. Here the space under the lower end of the valve stem is relieved to the atmosphere, so that the steam may not get beneath the stem and tend to lift the valve.

The mechanisms used for operating lift valves show a great variety in form. With Fig. 283, the exhaust valve is worked by a cam and lever. Cams of all kinds are used, and many ingenious linkage mechanisms have been devised. As already noted, these are very fully given in Leist's book on valve gears.

(*g*) THE ENGINE WITHOUT CRANK SHAFT. — In steam pumps and similar small engines which are described by this heading, special means of moving the valve must be adopted. It is not an effective scheme to make the piston operate its own valve through some sort of lever device, because the valve must make its wide and rapid movement at the time when the piston is moving very slowly, near the end of stroke. In the duplex steam pump, this difficulty is surmounted by letting each piston drive the valve which controls the other cylinder. The essential part of the machine is drawn in Fig. 285, the cross view at the right showing the valve levers or rockers. It will be noted that lever 1 is direct, while lever 2 reverses the movement of the piston rod: they are both proportioned so as to give the same ratio of reduction of motion from piston to valve. The simple analysis of movement in

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Fig. 286 shows the working of this engine, and makes clear the need of levers of opposite character.

In this figure, A marks the right side, B the left side. For position I, with piston A just reversing at left end of stroke, valve B (driven by arm A) is likewise at the left, port B is wide open at the right end, and piston B is moving from right to left; but in order that piston A shall

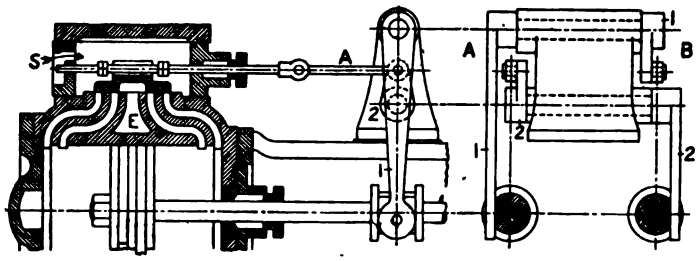

FIG. 285. — Valve Gear of Duplex Steam Pump.

presently move to the right, valve A must move to the right ahead of it, wherefore valve A must have a movement opposite to that of piston B. Following through the other three critical positions, and noting the directions of movement as indicated by arrows, we see that piston A leads, reaching any position half a stroke (one-quarter of the cycle) ahead of B.

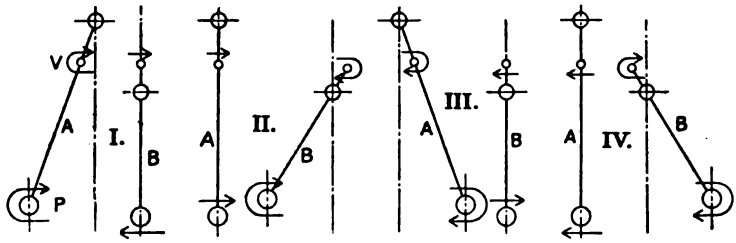

FIG. 286. — Valve Movement in the Duplex Pump.

Although moved by the pistons, the valves are not closely driven, but there is always some lost motion between the rock levers and the valves. In Fig. 285 this clearance or back-lash is permitted at the nuts on the valve rod, on each side of the valve; on larger pumps, the same effect, with easier access, is provided for at the outer end of the rod. With positive connection, the valves would move too soon, so that the pistons could not make full-length strokes. Valve setting in a pump like this consists in putting the pistons at mid-stroke and the valves in mid-position, and then making the rod clearances the same

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on both sides; minor adjustments may be found necessary after the pump is started, the object being to secure equality in the two strokes.

Figure 285 illustrates the fact that in engines of this class the valves are made with nearly or quite a zero lap. It shows also the use of two ports at each end, the outer one for live steam, the inner for exhaust. By thus having the exhaust port open into the cylinder at some distance from the head, a cushioning effect is secured, independently of the valve action, which will prevent the piston from striking the cylinder head. In order that this effect may be regulated at will (because less cushioning is needed at low than at high speeds), pumps of the larger sizes have stroke-regulating or cushion valves, arranged somewhat as shown by the sketch in Fig. 287. This valve varies the size of an opening through the wall between the two ports, permitting more or less exhaust by way of the outer part. Opening this valve lengthens the stroke, and *vice versa*.

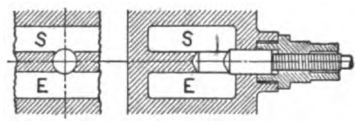

FIG. 287. — The "Dash-relief" Valve.

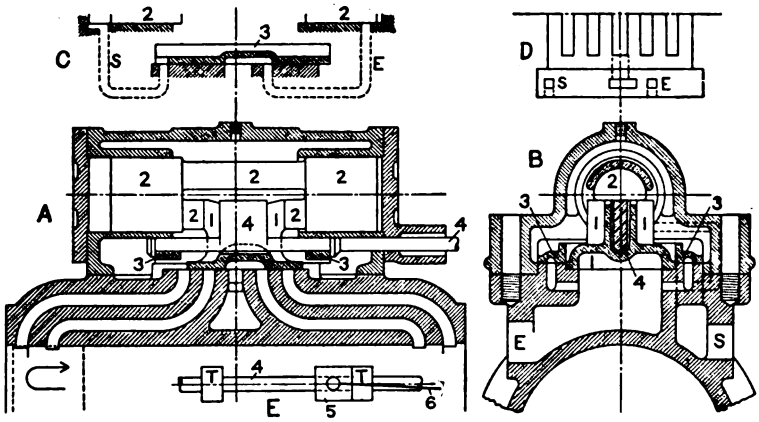

FIG. 288. — Valve Gear of the Deane Pump. 1. Main Valve; 2. Auxiliary piston which moves main valve; 3. Auxiliary valve, with ports shown in views C and D; 4. Valve rod; 5. Slide block; 6. Rod to pendulum lever.

(*h*) STEAM-ACTUATED VALVES. — The underlying principle of all single steam pumps is well exemplified by Fig. 288. The external gear consists of an indirect pendulum lever, equivalent to rocker 2 in Fig. 285, from which rod 6 (in view E) drives slide block 5. As the piston approaches the end of its stroke, block 5 strikes one of the tappets

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T, T, and moves valve 3. The latter, in form a hollow rectangle surrounding the main valve, serves two sets of little ports such as are diagrammed at C; through these steam is conducted to and from the cylinder spaces at the ends of the piston slide 2. Thus a small movement taken from the piston near the end of stroke sets into action what is really a complete secondary engine, which performs the function of moving the main valve. The reason for a double set of auxiliary ports is apparent when we note, in view C, that the steam port is carried clear to the end of the chamber, while the exhaust port is kept in from the head, so as to insure cushioning.

All gears of this type have essentially the same elements, but in the form and arrangement of the secondary valve, ports, and gear there is wide variety.

(*i*) INDICATOR DIAGRAMS FROM STEAM PUMPS. — These engines all take steam through the full stroke, and when working against the nearly constant resistance of water give an indicator diagram which

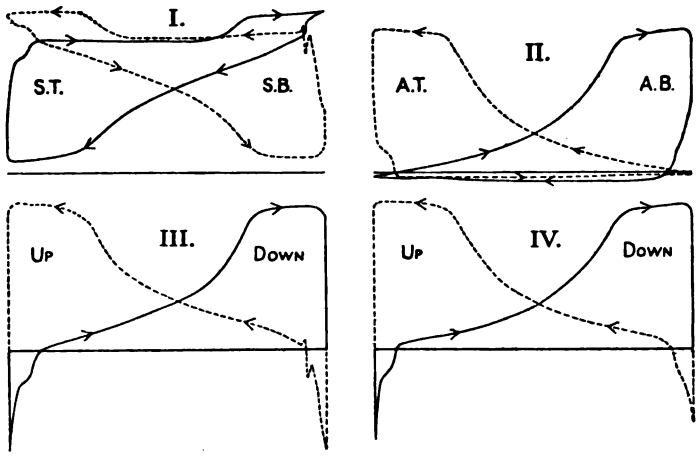

FIG. 289. — Pressure Diagrams from the Air-brake Pump.

approximates a rectangle — except as the reaction may be modified by the effect of inertia of a long column of water. When the resistance is not constant, but increases along the stroke as in the air-brake pump or in a condenser " air pump," the steam diagram takes the peculiar form at I in Fig. 289. Here I and II are corresponding indicator diagrams from steam end and air end, brought to the same scale of pressures. Considering simultaneous lines of forward pressure and back pressure in I, we see that a small driving force at first is obtained chiefly by throttling the exhaust; instead of taking place while the piston is at or

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near the end of its stroke, the release (with drop in pressure) is distributed along a good part of the return stroke. As the air resistance increases, the piston, which started off rapidly, slows down; and during the expulsion of the compressed air the two steam pressures are nearly constant at their maximum and minimum values. Diagrams of effective driving force and effective resistance are plotted at III and IV, from I and II, respectively, to show how very closely the two force actions agree, there being between them only the small differences needed to accelerate the pistons.

The necessary throttling action is secured in large degree by the use of very small ports: thus in the machine from which the diagrams in Fig. 289 were taken, the ratio of port area to piston area is about 1 to 130, as against perhaps 1 to 10 in an ordinary engine.

(*j*) THE SELF-CENTERING VALVE. — Fig. 290 is introduced with the purpose of illustrating a type of valve gear used in a wide variety of

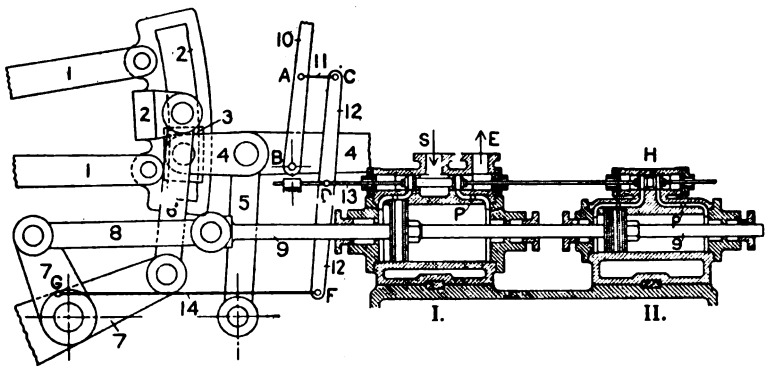

FIG. 290. — Reversing Gear for a Large Rolling-mill Engine. I. Operating cylinder, with steam; II. Brake and holding cylinder, with water or oil.

regulating devices, in which the part moved or adjusted must follow a control lever or handle. We are not now concerned with the Stephenson gear, pieces 1 to 7, but with the valve mechanism of the operating cylinder. The floating lever CDF is pivoted on the valve rod 13 at D. Suppose that hand lever 10 is swung to the left: CF will follow it, turning on F, and drawing the valves on rod 13 to the left. This will admit steam to drive the pistons to the right; and as 7 turns clockwise, it will push lever FC to the right, at F, compensating for the original displacement by hand, and bringing the valves back to the closed and holding position. Thus the main link 2 takes always a position definitely related to that of the hand lever 10. This principle has many applications, and the mechanisms which embody it differ greatly in form and detail.

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§ 45. Steam-engine Governors

(*a*) FUNCTIONS OF THE GOVERNOR. — The two types of cut-off governors have already been described, as to general form and working and as to their control of valve action, the fly-ball governor in § 43 (*d*), the shaft governor in § 2 (*m*) and § 39 (*b*). In a study of the forces which act within governors, two main questions come up, or two functions are to be considered. The first is the question of "regulation," or of the manner in which the speed in steady running varies with the load; the second is concerned with "adjustment," or with the behavior of the governor while in the act of accommodating the engine to a change of load: involved in both, and by no means of subsidiary importance, is the question whether the governor will hold steadily the position corresponding to a constant load, without yielding unduly to the action of secondary disturbing forces. By "close regulation" is meant that the whole range of load is covered with only a small change in speed — this change being normally a decrease as the load increases. An ideal governor, while steady under constant load, would respond at once to any change in the main controlling forces, following the load to the new position of equilibrium and stopping there, without superfluous movement on its own account. To realize this ideal is by no means an easy problem.

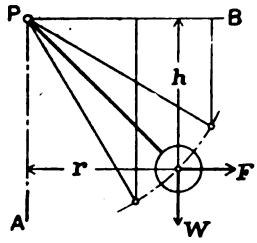

FIG. 291. — Element of the Fly-ball Governor.

(*b*) TYPICAL FORCE ACTION. — To establish certain fundamental concepts and principles, consider the simplified arrangement in Fig. 291. The ball at the end of the arm pivoted at P can swing in the plane of the drawing, while this plane has a rotary motion about the axis AP. For equilibrium, centrifugal force F must balance gravity force W, or there must be an equation of moments,

$$Fh = Wr. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (208)$$

The force F is a function of the r.p.m. N, according to the relation

$$F = \frac{W}{g}\frac{v^2}{r} = aWN^2r, \quad . \quad . \quad . \quad . \quad . \quad . \quad (209)$$

where a is the resultant constant. Substituting in the first equation and getting a new constant,

$$aWN^2rh = Wr, \quad \text{and} \quad N = b\sqrt{\frac{1}{h}}. \quad . \quad . \quad . \quad . \quad (210)$$

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This shows how the balancing speed varies with the position of the pendulum arm.

The equilibrium represented by Eq. (208) is a perfectly general requirement: but the transformations which follow belong to the particular case. If the pivot P is not on the axis of rotation of the governor, the moment arm of W about P is not the same as the radius which determines centrifugal force: then the relations and their expressions will become less simple than in Eq. (210).

(c) STABILITY. — To see how and why stability of position exists in a governor, return to Fig. 291 and suppose the rotation about the axis AP to be maintained at a constant speed N. If by some external force the ball is pulled down from its normal position, the moment of W, dependent upon r alone, will decrease more rapidly than that of F, which varies as the product rh; then the excess of the latter will swing the arm upward. With displacement on the other direction, r increases more rapidly than rh, and again there is an unbalanced moment acting toward the normal position. Roughly, with a constant speed, the force tending to return the ball to the neutral position is proportional to the displacement from that position.

For a series of governor positions and at a constant speed N, let values of the moment (Fh) be calculated and compared with (Wr); as there is less difference in their manner of variation, less change of speed will be needed to establish a new balancing position: the governor will be more nearly isochronous (of constant speed), but the stability against a disturbing force will be less. A perfectly isochronous centrifugal governor mechanism has no stability at all, and cannot be used to control an engine.

(d) ACTION IN ADJUSTMENT. — When the speed of an engine changes under change of load, the equilibrium position of the governor is shifted. The resulting unbalanced force at once begins to produce motion in the mechanism, but the acceleration of the mass requires a certain amount of time. Generally, the governor comes to the new position with some velocity and momentum, and swings past until checked by the reversed balancing force, then swings back in the same way. The tendency is to set up a regular pendulum motion, until the energy of the initial impulse has been absorbed in friction or in some other brake action. The study of this phase of the performance of governors is one of the most complicated subjects in the Mechanics of Machinery, and nothing but the preceding brief outline is appropriate here.

(e) REGULATION BY THE FLY-BALL GOVERNOR. — In Fig. 292 are outlined three typical forms of this governor, all weight-loaded, or with

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gravity as the force that acts against the centrifugal force of the balls: sometimes the load is furnished partly by springs, especially in small throttling governors. The first example in Fig. 292 corresponds with Fig. 268; it is slow-running and largely self-balanced, the weight on the central slide being small in comparison with that of the balls them-

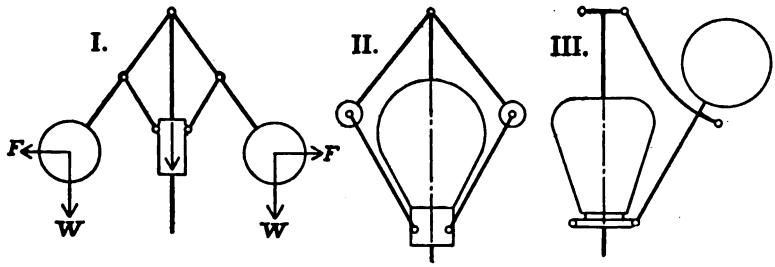

FIG. 292. — Types of the Fly-ball Governor: I. Common Low-speed Governor; II. High-speed Governor, Weight-loaded; III. The Proell Type.

selves. No. II runs at high rotary speed, hence the balls are small and most of the counterforce is furnished by the big central weight. No. III is a prominent German design: by placing the balls as shown, much closer regulation is secured than, for instance, in I.

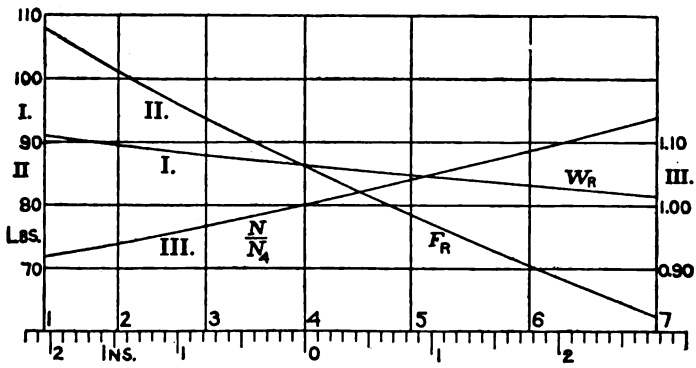

FIG. 293. — Curves of Regulation for Governor in Fig. 268.

To show how the general relations in Art. (*b*) work out for a particular case, results from the analysis of the governor in Fig. 268 are given in Fig. 293. The base of this diagram is the vertical path of a point on the central slide or sleeve, piece 6 in Fig. 268. The numbered ordinates correspond with a series of equally-spaced positions of the ball on its arc of travel; the ordinary range is from 2 (low) to 6 (high). By methods of kinematics, easy enough to apply but hardly appro-

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priate for description here, the forces of gravity on all the moving pieces of the governor, including the poise W, are reduced to the slide and there combined in the single resultant W_R. This tends to pull the governor downward and varies, with position, as shown by curve I. Similarly, all centrifugal forces are reduced to equivalent lifting effects upon the slide and combined in F_R, which varies with both position and speed. Curve II gives F_R as worked out for the speed which gives equilibrium at position 4, or for 86.2 r.p.m. in this mechanism and with the poise at the middle of its arm: by moving the poise from one extreme to the other, the mid-position speed can be changed from 79 to 93 r.p.m. of the governor. Comparing curves I and II, we see that to make F_R equal W_R, the speed must be lower at low positions, higher for the upper range. Curve III shows how N must vary for balance in the governor, representing the ratio of changing N to N_4 at mid-position.

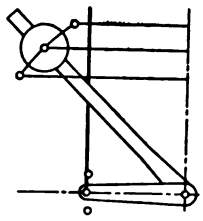

FIG. 294. — Counterpoise on Inclined Arm.

This governor is by no means a close regulator, the total variation from ordinate 2 (resting on stop ring) to ordinate 6 (highest running position) being about 15 per cent of the middle speed. Fig. 294 shows one simple scheme for improving the action of such a governor: by putting the poise W on an inclined arm, its moment arm will be made to decrease as the slide rises, and thus

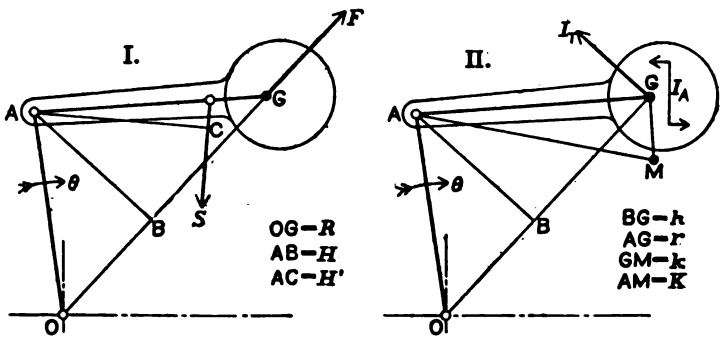

FIG. 295. — Force Action in the Shaft Governor.

curve I in Fig. 293 will be brought a little nearer to the inclination of curve II.

(*f*) FORCE ACTION IN THE SHAFT GOVERNOR. — In Fig. 295, the axis of rotation is projected in the point O, and OA is a radius in the plane of the wheel, out to the pivot pin A on which the weight arm AG can turn. At I is represented equilibrium between centrifugal force F,

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with moment arm AB, and spring force S, with arm AC. At II is shown a force action which comes into play during adjustment: the wheel being given an angular acceleration about O in the clockwise direction

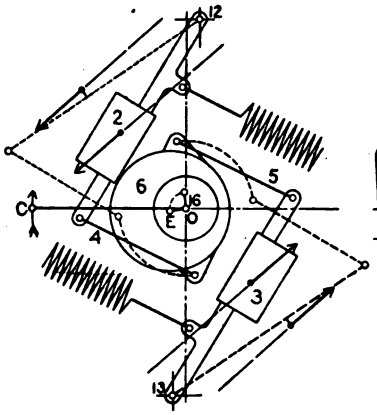

FIG. 296. — Buckeye Governor.

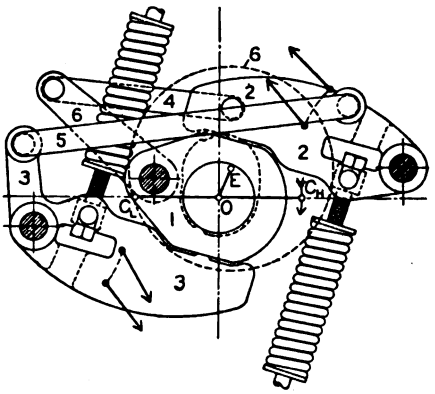

FIG. 297. — Westinghouse Governor.

the inertia of AG will tend to turn it outward about A, thus helping the unbalanced centrifugal force to bring the governor to its new posi-

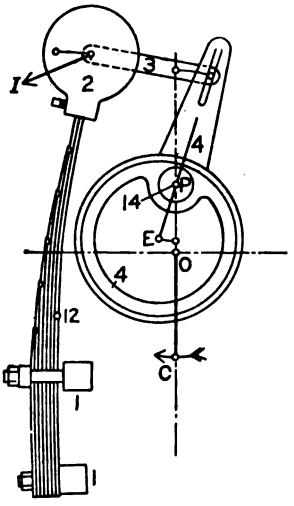

FIG. 298. — Robb-Armstrong Governor.

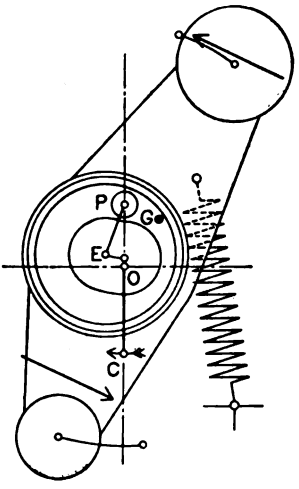

FIG. 299. — Rites Governor.

tion. Into the detail of this action, intended to be illustrated by some extra lines on the diagram, we shall not enter.

(*g*) TYPES OF SHAFT GOVERNOR. — Figs. 296 and 297 show two of

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the older governors, so arranged, in the relative position of weight and of arm pivot, that inertia can exert very little influence. Further, in having a practically symmetrical form with reference to the axis of the shaft, they are balanced against any possible disturbance by gravity force. Since the governor rotates in a vertical plane, the gravity pull on each mass constantly varies in relative direction; and while this force is comparatively small, its cyclical fluctuation tends in some degree to set up an oscillating motion.

Figures 298 and 299 represent governors in which inertia as an active influence during adjustment is intentionally given a prominent place, although both depend upon centrifugal force to determine the regulation. In these designs comparatively little effort is made to secure gravity balance, experience having shown that at high speeds the effect of unbalance is very small.

(*h*) CONTROL OF REGULATION. — In order to control the running speed and its manner of variation with load, most governors permit

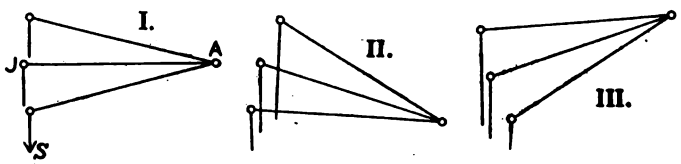

FIG. 300. — Types of Spring Connection.

change of the mass in the weight or ball, and change of spring tension; while some have provision for moving the point of spring connection, on the weight arm. Concerning the last matter, Fig. 300 shows different types of spring action. In the first case, the spring keeps an almost constant lever arm about A; in the second the arm decreases as the spring is stretched, while in the third it increases. The adoption of a particular point of spring attachment is commonly a question of original design rather than of subsequent adjustment; if provided for, the latter will probably take the form of a change of length in the arm AJ.

To add mass to the governor weight will make the engine run slower, to tighten the spring will speed it up. A combined change may keep the average r.p.m. nearly the same, but will affect the regulation. A given spring has a certain scale, or ratio of applied force to deflection produced. In the movement of the governor over its range, there is a definite change in spring deflection, hence a fixed increment of tension. If the initial tension is high, the increment bears to it a smaller ratio than if it is low. We see then that to increase mass and tension together will make the regulation closer, or the variation of speed less.

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In electrical service, it is quite usual to provide means for adjusting speed from the switchboard, so that units can readily be synchronized for parallel operation — the work being done by a little motor mounted on the governor.

(*i*) THE THROTTLING GOVERNOR. — The example in Fig. 301, in its mechanism for regulating speed, is a good representative of

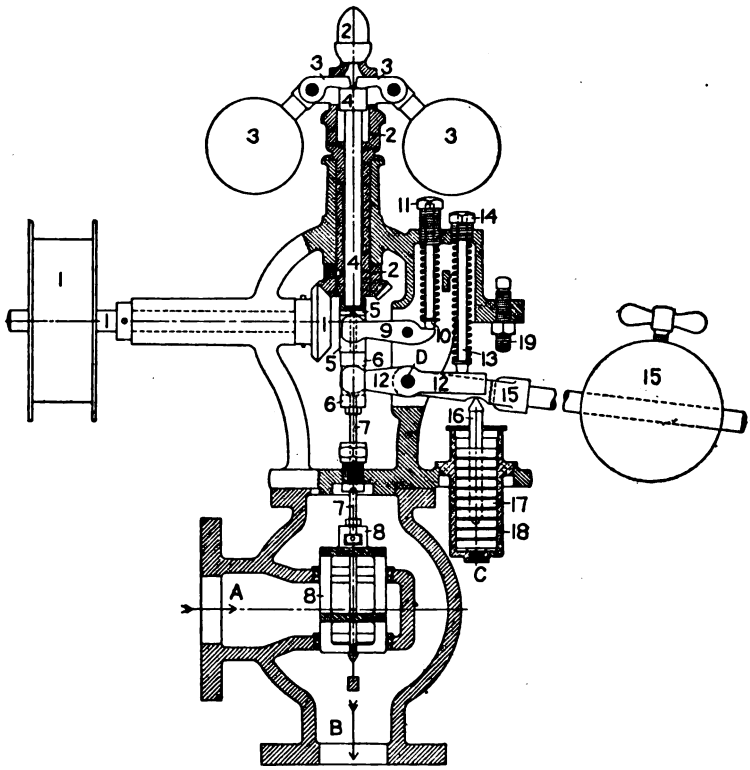

FIG. 301. — Gardner Governor for Air Compressor.

throttling governors in general: beside this, there is an apparatus for shutting down the engine when the air pressure rises to a certain desired height, and either device can operate independently of the other.

The main mechanism, consisting of pulley spindle 1, hollow governor spindle 2, and fly balls 3, 3, is self-evident. Through the rod 4 and the intermediate pieces 5 and 6, the valve 8 is pushed down when the balls fly out, throttling the steam. This valve, being of the piston rather than of the double-seated disc form, is perfectly balanced, so that the very light stem 7 is sufficient: and the crossbar below the

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valve insures that it shall never fall so far as to admit steam when it is intended to be closed. Piece 5 is held against 4 by the first spring, acting through lever 9; similarly, 6 is held against 5 by lever 12, the springs both having a certain share in determining the running speed. From the top of 6 a small pin or spindle projects into a central hole in 5, to keep these parts always in line.

The pressure governor consists of the cylinder 18, receiving the air pressure at C, with the plunger 17 and the weighted lever 15 which is pivoted at D. When the pressure gets to the desired maximum and raises the weight arm, the knife edge on the upper end of 16 pushes against the outer end of lever 12 and closes the valve. The stop screw 19 can be set so that steam will not be quite shut off by this action, for if the compressor is completely stopped, it may be in a position from which it will not start when steam is again admitted.

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CHAPTER IX

ACTION OF THE STEAM IN THE TURBINE

§ 46. Dynamics of Jet Action

(*a*) IMPULSE UPON THE JET. — The method developed and applied in § 16, for calculating the velocity of the steam jet, passes over all intermediate details of the process of accelerating the steam mass, and uses simply the final, overall relation, that since the pound of steam has received a certain amount of pressure work it must have gained an equal amount of kinetic energy.

In the formation of the jet, steam with a negligible velocity of approach is given a very high velocity, by a driving force which acts upon the steam mass in one direction and exerts an equal and opposite reaction upon the nozzle or containing vessel. This force, viewed preferably as a reaction, might be found by determining the variant pressures on the confining surfaces and summing up their components in the direction of the nozzle axis. Such an operation would be very complicated, whether by calculation or by experiment; but there is, fortunately, a simple way of getting at an equivalent net result.

Start with the fundamental dynamic relation, Force = Mass × Acceleration, or

$$F = MA, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (211)$$

and multiply both sides of the equation by time t; then

$$Ft = MAt = MV. \quad . \quad . \quad . \quad . \quad . \quad . \quad (212)$$

That is, when a free force F acts upon a mass M through the time t, and generates the velocity V from an initial state of rest, the product of force by time equals the product of mass by velocity, which latter is called the momentum of the moving mass. For applying these principles to the conditions represented in Fig. 302, let the fully-formed jet at the cross plane AB have the velocity V (feet per second), the cross area a (square feet), and the specific weight w (pounds per cubic foot). The discharge per second will be $W = waV$ pounds, and its mass

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$M = W/g$. Taking any time interval t, substitution in the general equation (212) gives

$$Ft = \frac{waVt}{g} V = \frac{WtV}{g}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (213)$$

or

$$F = \frac{waV^2}{g} = \frac{WV}{g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (214)$$

With a unit or one-pound-per-second jet, $W = 1$; then the actual diffused and variant pressure which produces the jet is equivalent to a single, concentrated force of the value

$$F = \frac{V}{g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (215)$$

As represented in Fig. 302, this may be called the impulse upon the jet: upon the nozzle there must be an equal and opposite reaction.

(*b*) Impulse and Reaction of the Jet. — In Fig. 303 a jet of steam is shown as flowing through a pipe or tube and impinging squarely upon a flat plate MN. The latter spreads the stream sidewise in all direc-

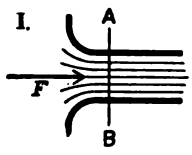

Fig. 302. — Impulse upon the Jet.

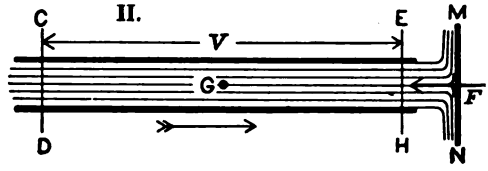

Fig. 303. — Impulse Exerted by the Jet.

tions, destroying the velocity in the original direction of flow. The pressure on the plate MN, equal and opposite to the force F, in the figure, which is required to hold this in position, is the impulse of the jet.

Figure 303 illustrates a derivation of the value of force F through work and energy relations, instead of momentum. The mass to be discharged in one second is shown as included between the planes CD and EH, which are separated by the distance V feet. At the beginning of the particular second under consideration, the plane EH just touches MN; and the center of mass G being then at the distance $\frac{1}{2} V$, this is the average distance through which the resistance F will act upon the mass W/g in bringing it to rest. Then equating work of retardation with kinetic energy lost, we have

$$F \times \frac{V}{2} = \frac{WV^2}{2g}, \quad F = \frac{WV}{g} \quad . \quad . \quad . \quad . \quad . \quad (216)$$

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Figure 304 shows the essential form of an apparatus which has been used in a number of experiments made to determine the reaction of a steam jet. The chamber represented by the square outline in I is at the lower end of an elastic tube, through which steam of high pressure has entrance. The steam blows out toward the right and tends to throw the box toward the left, exerting a reaction which can be measured by some weighing device or dynamometer. The arrangement is very effective as an illustration of jet action: but in its use for quantitative experiment there must be a clear understanding of the conditions existing. Suppose, for instance, that with a flow from a high pressure p_1 into a low pressure p_2, a plain converging nozzle (or an orifice with rounded entrance) is used, as in the first sketch. At the mouth or outlet of the nozzle, the steam in the jet will have the pressure $p_0 = 0.58\ p_1$

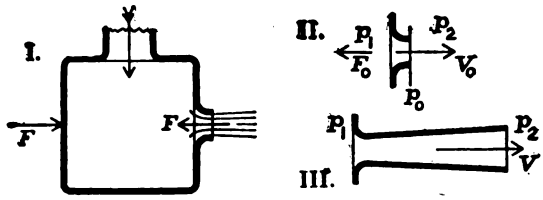

FIG. 304. — Reaction of Jet Formation.

and the velocity V_0 produced by expansion from p_1 to p_0 — note the representation at II, and refer to the remarks on this state of affairs in § 16 (*h*). Beside the dynamic reaction F_0, there will be then a static-pressure reaction in the same direction; because upon the area a_0 on the right side of the vessel (the nozzle mouth) reacts the pressure p_0, while upon the equal area on the other side acts only p_2. If the diverging nozzle at III is so proportioned as just to reduce the pressure in the jet at its mouth to p_2, all static pressures will be balanced and only the dynamic reaction will be measured. Note what is said in § 47 (*f*) concerning the case of excessive divergence of the nozzle.

EXAMPLE 42. — Steam initially dry-saturated flows from a pressure of 120 lb. abs. into 15 lb. abs., through a short converging nozzle with a least area of 0.25 sq. in. Assuming ideal flow, as defined in § 16 (*a*), find the discharge per second, the dynamic reaction, and the total reaction that would come upon the dynamometer in an apparatus such as is partly outlined in Fig. 304.

Take the pressure at the mouth of the nozzle or in the plane of the orifice to be $p_0 = 0.58\ p_1 = 69.6$ lb. From Table 7, page 118, the throat area of the unit jet from 120 lb. abs. is 0.5868 sq. in.; then through 0.25 sq. in. the flow will be $0.25 \div 0.5868 = 0.4261$ lb. per sec.

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By Table 6 directly, the ideal velocity at p_0 is 1474 ft. per sec. Substitution in Eq. (214) gives

$$F = \frac{0.4261 \times 1474}{32.16} = 19.53 \text{ lb.}$$

With a static pressure of 69.6 lb. in the steam at the plane of the orifice, there will be on the 0.25 sq. in. of outlet a force of 17.4 lb.: on the opposing 0.25 sq. in. at the other side of the box acts a force of $15 \times 0.25 = 3.75$ lb.: then the unbalanced steam pressure is 13.65 lb., which raises the total reaction to $R = 19.53 + 13.65 = 33.18$ lb.

EXAMPLE 43. — With the conditions in the last problem, find the proper mouth area of a diverging nozzle, then get the velocity and the dynamic reaction with full expansion in the nozzle.

With $p_1 = 120$, $p_2 = 15$, the pressure ratio is 0.125. From Table 6, the area ratio a/a_0 is 2.072, so that the mouth area should be $0.25 \times 2.072 = 0.518$ sq. in. To the area ratio 2.072 corresponds a diameter ratio of only $\sqrt{2.072} = 1.44$.

Again by Table 6, the velocity is 2755; and this in Eq. (214), with W from the last problem, makes

$$F = \frac{0.4261 \times 2755}{32.16} = 36.50 \text{ lb.} = R.$$

(c) DEFLECTION OF THE JET. — In Fig. 305 a flowing jet or stream is depicted as entering at A and leaving at B a frictionless channel of uniform curvature and cross section. The velocity remains constant

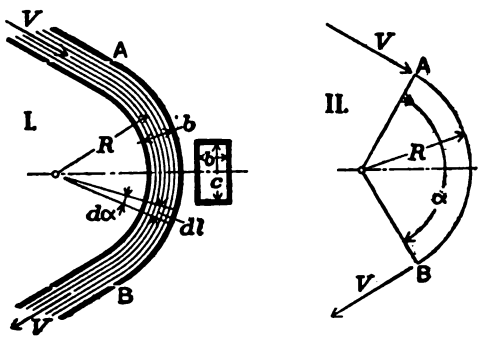

FIG. 305. — Deflection of a Jet.

in intensity, and the action involved in its continual deflection is that of simple transverse, centripetal acceleration, against which reacts the centrifugal force of the stream.

The essential dimensions and symbols are as follows:

a = area of cross section of channel, equals $b \times c$, the stream having the width b and the depth c;

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R = radius of center line of stream;
α = angle between any pair of radii;
l = length measured along curved center line;
v = specific volume of fluid in jet;
w = weight per cubic unit = $1/v$;
V = velocity of flow;
A = acceleration;
W = weight of fluid discharged per second;
F = impulse of jet.

Consider an element of the stream which is included between two radial planes at the angular distance $d\alpha$ from each other:

$$\text{Length} = Rd\alpha; \quad \text{volume} = aRd\alpha; \quad \text{mass} = \frac{waRd\alpha}{g} = m.$$

The centrifugal force of this element is mV^2/R, or

$$f_c = \frac{waRd\alpha}{g}\frac{V^2}{R} = \frac{waV^2}{g}d\alpha = Fd\alpha, \quad . \quad . \quad . \quad . \quad (217)$$

the reaction F being substituted from Eq. (214). The important result follows that the total pressure required to deflect a jet through an angle α (this pressure being distributed along the curved guiding surface, and found by integrating from 0 to α as indicated at II) is equal to the impulse of the jet multiplied by the value of α in radian measure.

The disappearance of R from the expression for f_c in Eq. (217) shows that with a given jet, or with V, W, and a fixed, the total deflecting pressure, or the summed up reaction of the jet against the curved surface, is independent of the radius of curvature. With a certain angle α, to make R bigger will make the channel longer and the contained mass of steam greater, and this compensates for the inverse variation of unit force with R.

In Fig. 305, let us take the outer surface of the element of volume to be $cRd\alpha$, disregarding the fact that the outer radius is really $(R+\frac{1}{2}b)$; division into the force f_c will give the unit centrifugal pressure

$$p_c = \frac{wbcRd\alpha}{gcRd\alpha}\frac{V^2}{R} = \frac{wb}{g}\frac{V^2}{R}. \quad . \quad . \quad . \quad . \quad . \quad (218)$$

EXAMPLE 44. — Using, directly or indirectly, values in the 0.40 line of Table 6, let a steam jet at 48 lb. abs. pressure, with the velocity 1891 ft. per sec. and the specific volume 8.33 cu. ft. flow in a channel 0.5 in. wide and 1 in. deep, having the mean radius 2 in. What is the impulse of the jet and what the unit centrifugal pressure on the guiding surface, assuming that no losses by friction, etc., need be considered?

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Here $a = 0.5$ sq. in. $= 0.00347$ sq. ft.

The rate of flow is $W = \dfrac{0.00347 \times 1891}{8.33} = 0.788$ lb. per sec.

The impulse is $F = \dfrac{0.788 \times 1891}{32.16} = 46.3$ lb.

In Eq. (217), b, c, and R must be reduced to feet, so that

$$f_c = \frac{w}{g}\frac{bc}{144}\frac{Rd\alpha}{12}\frac{12\,V^2}{R} = \frac{wbcRd\alpha}{144\,g}\frac{V^2}{R}; \quad \ldots\ldots \quad (217')$$

but to get p_c per sq. in., area $cRd\alpha$ is used without change; then Eq. (218) becomes

$$p_c = \frac{wb}{144\,g}\frac{V^2}{R}. \quad \ldots\ldots\ldots\ldots \quad (218')$$

For this problem, noting that $w = 1/8.33$, we have

$$p_c = \frac{0.5 \times 1891 \times 1891}{8.33 \times 144 \times 32.16 \times 2} = 23.17 \text{ lb. per sq. in.}$$

Using the larger outer surface of the channel, with the radius 2¼ instead of 2 in. changes this to

$$23.2 \times \frac{2}{2.25} = 20.6 \text{ lb.}$$

Evidently, there will be a strong crowding of the stream against the guiding surface, with some consequent departure from the ideal, orderly flow which has thus far been assumed.

(*d*) Action of Jet upon Vane. — In Fig. 306, let AB represent the full velocity V of the jet as delivered by the nozzle. This jet flows upon a vane or into a "bucket" CD, which has the velocity V_T. To get the relative velocity V_1 of entrance, that of the nozzle relative to the vane, laid off at BC as V_T reversed, must be combined with V in the triangle ABC.

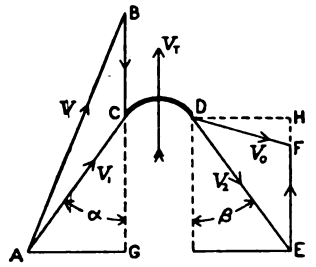

Fig. 306. — Velocity Diagram.

The function of the curved vane CD, or of the channel between two successive vanes, is to change the direction of the steam current so that it escapes with the relative velocity V_2, here equal to V_1 and making the same angle with the line of bucket movement. The dynamic pressure of the steam upon the curved guide, due to the inertia with which the current resists transverse acceleration, is the driving force in the turbine.

Combining with V_2 the vane velocity V_T at EF, we get DF or V_0 as

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the absolute exit velocity of the steam. The decrease in kinetic jet energy which accompanies the drop from V to V_0 is equal to the work done by the steam in driving the turbine wheel — except that a portion of this kinetic energy may have been wasted in friction and eddies, and be carried as a part of the heat content of the steam.

(*e*) DRIVING FORCE ON THE VANE. — Knowing the centrifugal force exerted by the jet upon a curved vane surface, the next step is to find the resultant, in a certain direction, of this radial force distributed along the vane. The problem is illustrated in Fig. 307, where the tangential force F_T is the resultant sought, acting in the direction of motion of the

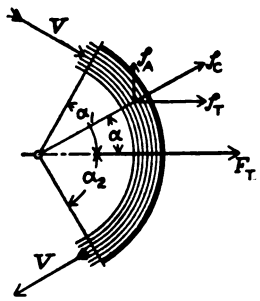

FIG. 307. — Resolution of Centrifugal Force.

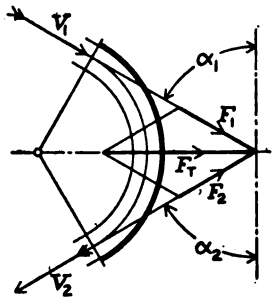

FIG. 308. — Combining the Impulses.

vane or bucket. Resolving any elementary force f_c, we get the driving component $f_T = f_c \cos \alpha$, and the axial component $f_A = f_c \sin \alpha$, the latter being parallel to the axis in the usual type of axial-flow turbines. To get F_T we make two integrations, one on each side of this resultant line. From Eq. (217) we have

$$f_T = \frac{waV^2}{g} \cos \alpha d\alpha; \quad \ldots \ldots \ldots \quad (219)$$

then, for the resultant force,

$$F_T = F\left(\int_0^{\alpha_1} \cos \alpha d\alpha + \int_0^{\alpha_1} \cos \alpha d\alpha\right)$$

$$= F(\sin \alpha_1 + \sin \alpha_2), \quad \ldots \ldots \ldots \quad (220)$$

F being the impulse of the jet, as appears from Eq. (214).

For the resultant axial force F_A, a similar deduction gives

$$F_A = F(\cos \alpha_1 - \cos \alpha_2); \quad \ldots \ldots \quad (221)$$

obviously, it is highly desirable that this force be made zero, since otherwise there would be a dynamic end thrust on the rotor.

The form of Eq. (220) suggests at once the simple method set forth in Fig. 308. Considering the jet as exerting a positive impulse F_1 at

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entrance and a negative impulse or reaction F_2 at exit, we have only to combine these forces, or their rectangular components, to get F_T and F_A. Since impulse is proportional to velocity, this gives a very easy and convenient method for the solution of problems in force action on the vanes of a turbine.

(*f*) Types of Vane Action. — The meaning of the terms impulse and reaction, as used to distinguish the manner of working of steam turbines, has been explained in a general way in § 4. The turbine elements shown at I in Figs. 309 and 310 are intended to illustrate a fuller definition. The velocity diagrams at II are made a little more compact than in Fig. 306, by omitting the vane profile and drawing V_2 in sequence with V_1. The various forces in the diagrams at III are parallel and proportional to the corresponding velocities.

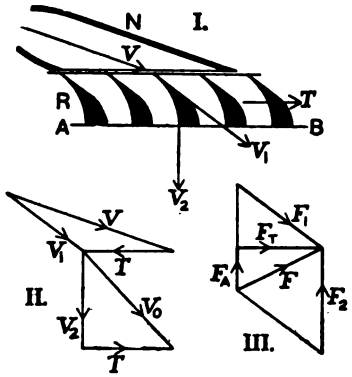

Fig. 309. — Driving by Impulse Only.

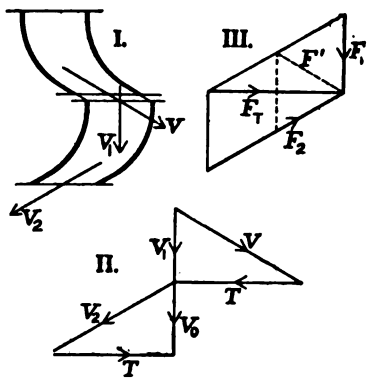

Fig. 310. — Driving by Reaction Only.

In Fig. 309, the half vanes are so formed that the steam leaves them in a direction perpendicular to the line of motion: then only the impulse at entrance is effective to produce driving force F_T, and the energy abstraction is very imperfect, as appears from the large residual velocity V_0. Another way of expressing this last condition is to say that the resultant F and the working force F_T are comparatively small because F_1 and F_2 have such a large angle between them. Very evidently this type of element must be completed by making the vanes symmetrical with respect to the center line AB, thereby getting the well-known form shown in Figs. 16, 21, etc. In other words, what is called an impulse turbine really is driven equally by impulse and by reaction.

But while the scheme outlined in Fig. 309 is not effective, that of Fig. 310 is entirely so. Receiving the steam normally (to the line of motion) with the velocity V_1, the reaction element accelerates it to V_2

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and discharges it at a wide angle from the normal. This leads to the force diagram III, where we see that the resultant F or F_T is large and is right along the line of movement. It appears then that while the "impulse" turbine must use reaction, the "reaction" turbine can get along without impulse. The real distinction lies in the fact that in one case velocity is generated wholly in the nozzles, in the other case in both fixed and moving vanes. The characteristic vane profiles shown in Figs. 311 and 313 result from this underlying difference. With a full understanding of what lies back of the terms, there can be no objection to the ordinary nomenclature: and a further distinction can be drawn in that the effective exit reaction in an impulse turbine is due wholly to deflection of the jet, while that in the reaction turbine is due to both deflection and linear acceleration. The dotted line F' in Fig. 310 III shows what the resultant would be with F_2 equal to F_1.

After the explanation just given, the actual impulse-element profiles and their diagrams of velocity and of force in Figs. 311 and 312 ought to be self-explanatory. Of the wheel with buckets formed in the rim,

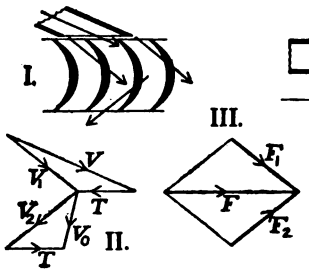

FIG. 311. — The Radial Vane, with Side Admission.

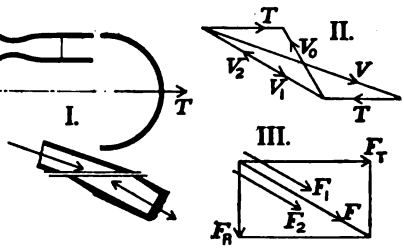

FIG. 312. — Bucket in Rim of Wheel, with Tangential Admission.

no examples have been given in Chapter I; fuller illustration will be found in Fig. 354. In Fig. 312 the plane of diagrams II and III is that of the lower, sectional view at I, or it is the plane, perpendicular to the axis, in which the wheel rotates. Then the velocity diagram doubles back on itself, because the projection of V_2 is simply V_1 reversed. It is necessary, of course, that V_0 approach the radial direction, since the steam must escape from the wheel in that direction. The forces F_1, F_2, and F should really lie on the same line in their diagram, but are here separated for clearness of representation. Whereas with side admission the resultant F is itself the driving force, the same thing as F_T, with tangential admission there is a non-effective component F_R, acting to cause pressure in the bearings, but which can easily be balanced by admitting steam to diametrically opposite parts of the wheel.

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The reaction vanes in Fig. 313 differ from the limiting form in Fig. 310 in that they provide for a small effective impulse at entrance. Since it is always desirable that the peripheral velocity be no greater than is absolutely necessary, this T is usually made quite a little less than the

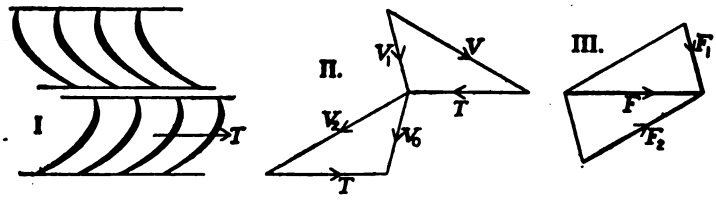

FIG. 313. — Diagrams for the Reaction Turbine.

projection of V, giving V_1 the slant shown. Then the vane profiles are made to fit the velocity diagram, as in any case.

(*g*) WORK ON THE VANES. — The effective dynamic driving force F (called F_T in the preceding discussion) acts upon vanes which have the velocity T; then the power, or rate of work performance, is

$$P = FT \text{ ft. lb. per sec.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (222)$$

We shall now apply to several typical cases the principle represented by this equation, still adhering to the primary assumption that there are no losses by friction or by other secondary actions. In the figures immediately following, the velocity diagrams (from which the impulsive forces may be directly determined) are changed to a rather more compact form than that used heretofore.

In Fig. 314, for instance, all the steam velocities are laid out from the point A as an origin: AB is the initial absolute velocity V, AC the

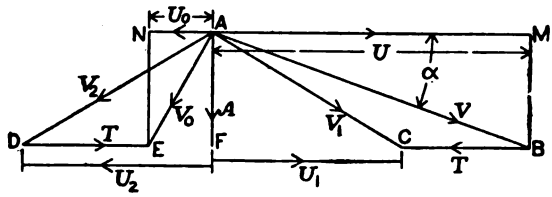

FIG. 314. — Velocity Diagram for the Impulse Element.

relative entrance velocity V_1; the exit velocity AD or V_2 is V_1 reversed symmetrically; and AE is the final absolute velocity, or V_0. For a discussion of work performance we are concerned, however, not with any total velocity V so much as with its component U in the direction of motion, or along T; the effective impulses at entrance and exit being

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proportional to the velocities U_1 and U_2, according to Eq. (214). With the relation

$$V^2 = U^2 + A^2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (223)$$

and with the axial component A remaining constant throughout the successive transformations which take place, we see that changes in the kinetic energy of the steam current are represented and measured by changes in the value of U^2.

Now for the impulse turbine, as represented by Fig. 314, and with the several velocities as there designated, the fundamental expressions are

$$U_1 = U - T \quad \text{and} \quad U_2 = U_1, \quad . \quad . \quad . \quad . \quad (224)$$

the latter equation embodying the condition of symmetrical reversal, and being subject to modification in the actual case. The effective impulses are now

$$F_1 = \frac{W}{g} U_1 \quad \text{and} \quad F_2 = \frac{W}{g} U_2,$$

and the work rate is

$$P = \frac{W}{g} T (U_1 + U_2) = 2 \frac{W}{g} (UT - T^2). \quad . \quad . \quad (225)$$

Examining this for the maximum value of P, in the usual manner, we get

$$\frac{dP}{dT} = 2 \frac{W}{g} (U - 2T), \quad . \quad . \quad . \quad . \quad . \quad . \quad (226)$$

which becomes zero when $T = \frac{1}{2} U$, thus proving the oft-stated principle that the velocity of the vanes should be one-half the effective

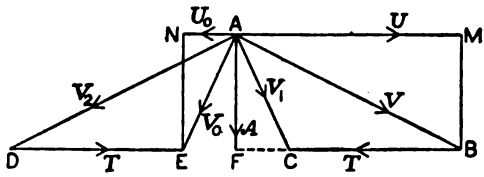

FIG. 315. — Velocity Diagram for the Reaction Element.

velocity of the steam jet for maximum efficiency. The greatest work rate is now

$$P_M = 2 \frac{W}{g} \times \frac{1}{4} U^2 = \frac{1}{2} \frac{W}{g} U^2; \quad . \quad . \quad . \quad . \quad . \quad (227)$$

or, as it should be, the full kinetic energy available in the weight W of steam that passes in one second.

(*h*) WORK IN THE REACTION TURBINE. — From the similar repre-

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sentation of the velocities in a reaction-turbine element, in Fig. 315 which corresponds with Fig. 313, we get the relations

$$U_1 = U - T, \qquad U_2 = U, \quad . \quad . \quad . \quad . \quad . \quad . \quad (228)$$

the second implying similarity between fixed and moving vanes. Then

$$F_1 + F_2 = \frac{W}{g}(2\,U - T),$$

and

$$P = \frac{W}{g}(2\,UT - T^2). \quad . \quad . \quad . \quad . \quad . \quad (229)$$

This is greatest when $U = T$, in which case the effective driving force is wholly due to reaction, since U_1 will be zero or the steam will enter the vanes at right angles to T, as in Fig. 310. The maximum value of P is now

$$P_M = \frac{W}{g}U^2; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (230)$$

but here U^2 represents only half of the energy of one complete stage, since U is generated twice, first in the fixed vanes, again in the moving vanes. The total kinetic energy generated and absorbed is properly expressed by

$$P_M = \frac{1}{2}\frac{W}{g}(2\,U^2), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (231)$$

this $(2\,U^2)$ being equivalent to the U^2 in Eq. (227).

The relations just deduced by mathematical reasoning call for fuller explanation. As implied, the stage embraces two rows of vanes, one fixed, the other moving. The function of the fixed element is to give to the steam the velocity and the kinetic energy which it must have if it is simply to move with the other element; then follows the question of change of motion within and relative to the moving element. This separation of the whole pressure drop and velocity generation into two parts is an important and essential feature of the turbine. In contrast with it consider the reaction "wheel," reproduced from Fig. 17 as Fig. 316. Within the hollow arm and at entrance to the nozzle, the steam mass has the kinetic energy due to the velocity T of the nozzle, and is at the full initial pressure p_1 (plus an increment due to centrifugal force). The pressure drop generates a relative velocity V, and the residual velocity of the steam after discharge is $V_0 = (V - T)$.

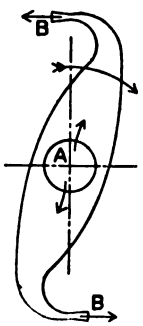

FIG. 316. — Outline of the Reaction Wheel.

This residual velocity cannot possibly be reduced to zero: for (with

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M representing the mass of steam discharged per second) reaction $F = MV$ and work rate $P = MVT$; then to make $T = V$ would make $P = MV^2$, whereas the whole available energy of the jet is only $W = \frac{1}{2} MV^2$. A closer analysis would show how the acceleration of the steam within the arm, as it flows outward, absorbs work. But without going into detail, we have the outstanding principle that if all of the pressure drop in a stage is reserved to the moving element, it will give to the steam a higher (relative) velocity than the (moving) nozzle is capable of assuming; hence there must be a considerable residual velocity, and even without secondary losses the turbine cannot reach unit efficiency in energy absorption.

It is of interest to note that, for a stage of given value, the vanes in a reaction turbine must move faster than in an impulse turbine. With a single expansion, the energy available will produce the effective velocity U_1, and the best speed of impulse vanes will be $\frac{1}{2} U_1$. In each of the two half-expansions of a reaction stage, the velocity generated will be $U_R = 0.707\, U_1$, but the best speed is now this full U_R. In application, the reaction turbine has always a larger number of stages for a given range of pressure.

EXAMPLE 45. — In Fig. 314, let the angle α be 20°, so that $\sin \alpha = 0.342$, $\cos \alpha = 0.940$. Making no allowance for secondary losses, calculate the velocities, forces, and work quantities in an expansion stage from $p_1 = 120$ lb. to $p_2 = 72$ lb. abs., with one pound per second of initially dry steam and with $T = \frac{5}{8} U$.

From Table 6, with $R_p = 0.60$, energy $E = 40.80$ B.t.u. or 31,740 ft. lb., and velocity $V = 1429$ ft. per sec.: then $U = V \cos \alpha = 1343$, $T = 0.375 \times 1343 = 503$, and $A = V \sin \alpha = 488$ ft. per sec. Assuming $V_1 = V_2$ and component A constant, as in Fig. 314, the driving force of this unit jet upon the vanes is

$$F = \frac{2\,(U - T)}{g} = \frac{5}{4} \frac{U}{g} = 52.2 \text{ lb.}$$

Acting on vanes which move with the velocity $T = 503$ ft. per sec., this force does work at the rate

$$P = 52.2 \times 503 = 26{,}260 \text{ ft. lb. per sec.,}$$

equivalent to $26{,}260 \div 550 = 47.8$ horse-power.

With the given values of U and T, the residual component $U_0 = \frac{1}{8} U = 336$ ft. per sec. Then

$$V_0^2 = A^2 + U_0^2 = 238{,}100 + 112{,}900 = 351{,}000;$$

and division by $2g$ reduces this (still for the one-pound-per-second jet) to the residual kinetic energy,

$$E_0 = 354{,}000 \div 64.32 = 5457 \text{ ft. lb.}$$

By direct subtraction of P from E,

$$E_0 = 31{,}740 - 26{,}260 = 5480 \text{ ft. lb.}$$

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Insufficient precision in the velocity values will account for the small discrepancy.

The efficiency of the turbine wheel in absorbing energy, under the assumed ideal conditions, would be

$$e = \frac{P}{E} = \frac{26{,}260}{31{,}740} = 0.828.$$

Actually, the efficiency of this high-pressure stage, measured in terms of net work deliverable by the turbine shaft relative to steam energy E, would probably lie below 0.50.

(*i*) VARIATION IN RUNNING SPEED. — It will be noted that neither Fig. 314 nor Fig. 315 is drawn for the case of ideal maximum efficiency, but rather for the conditions likely to be found in practice, where the vane speed T is made as low as is consistent with reasonably good working. To see the effect of thus lowering T from the ideal value, we discuss Eqs. (225) and (229) as follows:

For the impulse turbine $\quad P = 2\frac{W}{g}(U - T)\,T;$ For the reaction turbine $\quad P = \frac{W}{g}(2\,U - T)\,T.$

In both cases, let $T = nU$, and put the expression into the form, kinetic energy $\times$ a function of n; this gives

$$P = \left(\frac{1}{2}\frac{W}{g}U^2\right)(1 - n)\,4\,n; \quad P = \left(\frac{W}{g}U^2\right)(2 - n)\,n. \quad . \quad (232)$$

For the first case, let $m_1 = 4\,n_1\,(1 - n_1)$, and evaluate for T varying by twentieths from zero to $\frac{1}{2}\,U$; for the second, let $m_2 = n_2\,(2 - n_2)$, and go by tenths from zero to U. The common results are given in the tabulation below, and it is evident that the vane velocity T may be lowered to 70 or 75 per cent of the ideal value without serious loss of effect.

$n_1 = 0$	0.05	0.1	0.15	0.2	0.25	0.3	0.35	0.4	0.45	0.5
$n_2 = 0$	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
$m_1 = m_2 = 0$	0.19	0.36	0.51	0.64	0.75	0.84	0.91	0.96	0.99	1.0

An important assumption underlying this table is that the vanes be changed in form, with the running speed, so as to get full effect from the conditions existing in any case: and this leads to the next matter to be taken up.

(*j*) VANE FORM AND SPEED CHANGE. — The proper function of a set of curved vanes or guides in a turbine is, to receive a current of steam without shock, to change its direction without the formation of eddies, and to discharge it in a desired direction. The simplest question involved in the performance of this function will now be considered.

The elementary problem of accommodating the shape of the vane

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to a proposed velocity diagram, is illustrated by Fig. 317. In Case I the conditions are those for maximum efficiency in an impulse turbine, the vane velocity T being half of the effective entrance velocity U: the concave profiles are arcs of circles, made tangent to the lines of direction of the relative velocities V_1 and V_2; the convex profile is made up of a smaller arc and two tangents. If the channel between two vanes is to

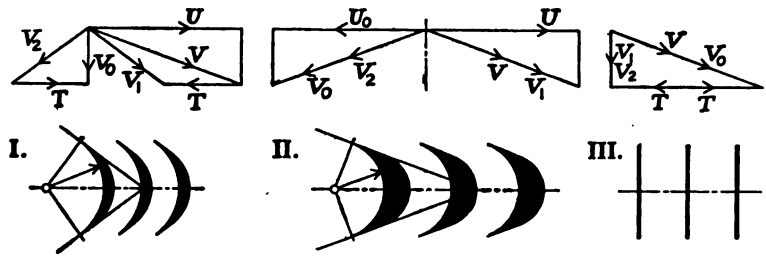

FIG. 317. — Vanes to Fit Various Speeds.

have an approximately constant width, the vane must, of course, be thickened toward the middle — see Fig. 319. This effect is much exaggerated at II, where the vanes are proportioned so as to receive and symmetrically reverse the full entrance velocity, when $T = 0$ or the vanes are standing still. Case III is at the other limit, when $T = U$ and the current simply flows between straight vanes, without exerting any driving force — this being the greatest speed at which the turbine could possibly be made to run by steam action upon symmetrical vanes. Change in the vane speed T, after the form of the vane has been fixed, is discussed in § 48 (*i*).

(*k*) CHANNEL FORM AND AREA OF CROSS SECTION. — In the most usual type of turbine, that with axial flow and having vanes with side admission, the channels for the passage of steam all lie within an annular space, which changes in diameter and in radial depth according to the requirement for effective area. The steam current has, at any critical point, an actual velocity V_n and a progressive component velocity A (see Figs. 314, 315, etc.): the latter is parallel to the axis or normal to the line of vane movement, but the total velocity, as also the direction of the channel, is oblique to these rectangular reference lines.

To understand the effect of this obliquity, consider Fig. 318, where the vanes in the rotor R are made straight, continuing the slant which actual vanes would have at entrance — being fitted, of course, to the velocity diagram II. Letting b_0 represent the pitch of the vanes or the width of the channel in the circumferential direction, we see that the effective width is much less, having the value

$$b = b_0 \sin a \quad \text{or} \quad b_1 = b_0 \sin a_1: \quad . \quad . \quad . \quad . \quad (233)$$

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in other words, making the walls helical decreases the width of the channel by an amount which increases with the inclination from the axial direction.

In the turbine, the relation between the steam velocities, as shown at II in Fig. 318, is the same as that between the channel widths (in inverse order) for

$$V = \frac{A}{\sin a} \quad \text{and} \quad V_1 = \frac{A}{\sin a_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (234)$$

Consequently the flow capacities are the same, since

$$Vb = V_1 b_1 = Ab_0. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (235)$$

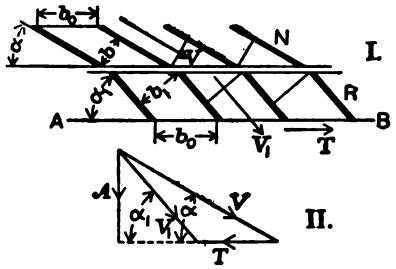

FIG. 318. — The Oblique Channel.

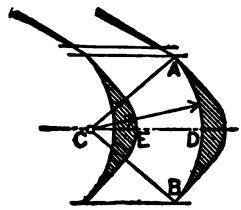

FIG. 319. — Channel of Constant Width.

The last expression, Ab_0, is then the criterion by which to measure the effective area for the passage of steam.

The preceding discussion determines relations which in the actual case exist at entrance to and exit from the channels between curved vanes. For the body of such a channel the conditions which give constant width are set forth in Fig. 319, where the curved portion of the passage is included between the radial lines CA and CB, the same center C being used for the arcs through D and E. The straight lines (tangents) which form a large part of the outer profile of the vane section have, of course, the inclination of the velocity V_1 as in Fig. 318, making the angle a_1 with the line CD.

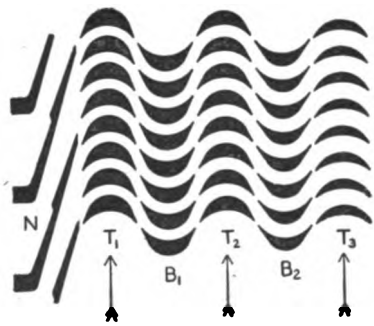

FIG. 320. — Element of Curtis Turbine.

An interesting conclusion from Eq. (235) is that if in an arrangement like Fig. 320 the vanes are brought to a sharp edge and formed as in Fig. 319, the diminution of obliquity will compensate for the diminution

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of velocity due to abstraction of energy — this on the assumption that the velocity changes are according to a diagram like Fig. 322, where A remains constant. Only for the purpose of accommodating extra losses of progressive velocity, as by friction and eddies, need the channels be increased in radial depth within the pressure stage.

(*l*) ACTION IN THE MULTIPLE-IMPULSE TURBINE. — Taking Fig. 320 as a typical example, we have its velocity diagram in Fig. 321, laid out

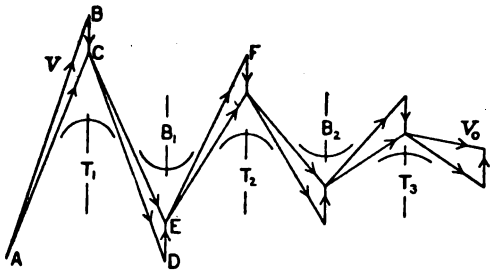

FIG. 321. — Multiple-step Velocity Diagram.

with the triangles in sequence, and showing how the bucket profiles are fitted to the lines of velocity direction. For the discussion of work relations, the single-pole diagram in Fig. 322 is preferable. Here the relative velocities of entrance and of exit are marked by the numbers 1, 2, and 3 for each set of vanes. Following the method of Art. (*g*),

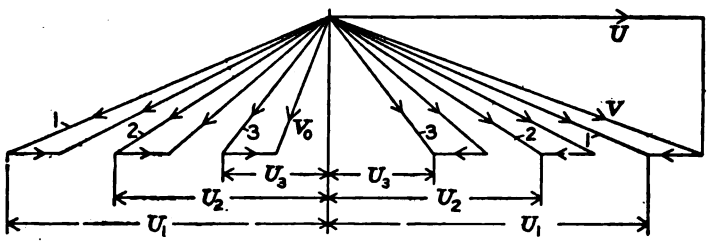

FIG. 322. — Velocity Diagram for a Three-impulse Stage.

and dropping for the time the factor W/g in Eq. (214), we have the following expressions for the driving forces in the respective stages:

$$F_1 = 2\,(U - T); F_2 = 2\,(U - 3\,T); F_3 = 2\,(U - 5\,T). \qquad (236)$$

Of course, the velocity factor T in $P_n = F_n T$ is the same all through, so that the work rates are proportional to the impulsive forces. If, for instance, T is one-seventh of U, as in the figure, the quantities of work done by the steam upon the vanes are as 6, 4, and 2 in the three velocity stages.

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To show the identity, under the conditions of simple theory, of the above example with the equivalent single-impulse element, we first add the three driving forces in Eq. (236), then multiply by T to get the power developed, the results being

$$F = 6\,(U - 3\,T);\ P = 2\,(U - 3\,T) \times 3\,T.$$

Since a single-impulse stage with the same limits V and V_0 as in Fig. 322 would have the vane speed $T' = 3\,T$, the desired equivalence is self-evident.

(*m*) PATH OF THE JET. — The absolute path of an element of the steam current, as it passes along the moving vane, is a matter of interest; and, in cases like Figs. 365 and 354 III and IV, is of considerable practical importance. Plotting this path is a simple geometrical process, which is shown in Fig. 323 for the side-admission vane.

First of all, the vane profile ABC is divided into, say, eight equal parts; then, from either velocity triangle, the distance that the vane will move on account of T while the steam travels over one interval with the velocity V_1 or V_2 is found. This is laid off parallel to BD, the proper number of times from each numbered point on ABC, and the result is the path ADE, which is tangent to V and V_0 at A and E respectively. For a tangential bucket, as in Fig. 354, the method would be essentially the same. In the reaction turbine it would be necessary to know how the steam is accelerated within the bucket; but with full peripheral admission there is no need of this determination.

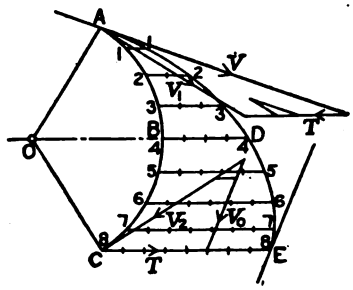

FIG. 323. — Plotting the Path of the Steam Current.

(*n*) ABSOLUTE VELOCITY IN THE VANE CHANNEL. — For the vane profile drawn in Fig. 323 and with the initial and terminal velocities there shown, the variant absolute velocity of the flowing steam is determined in Fig. 324. The vane velocity T is first laid out as AB, then from B is struck an arc with the radius V_1 or V_2: the length of this radius represents the uniform relative velocity assumed to exist along the channel, while successive angular positions show the direction of that velocity at the successive numbered points along the curve in Fig. 323. After putting in V_1 and V_2 and dividing the arc C8, we draw vectors from A to the points on this curve, and thus get a series of absolute velocities, referred to the fixed body of the machine.

At II, on a base which represents the vane profile when "developed,"

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the steam velocities are laid out as parallel ordinates. The kinetic energy of an element of the current varies as the square of this ordinate, and thus can be seen how the initial energy is gradually diminished as

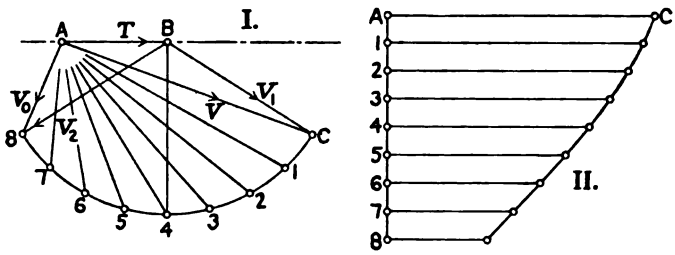

FIG. 324. — Determination of Absolute Steam Velocity.

the advancing jet performs work upon the moving vane. The experimental observations of pressure described in § 47 (*n*) indicate that this smooth and simple transformation of energy falls far short of realization.

§ 47. Experiments on the Steam Jet

(*a*) FLOW THROUGH ORIFICES. — A large number of experiments have been made to determine the rate of flow of steam through orifices and nozzles. The various results are generally quite consistent, and are put into the most useful form when brought into comparison with the calculated flow under ideal conditions, so that coefficients of discharge can be found. Several typical and representative sets of flow tests will now be given, and others will follow in the account of experiments where the rate of flow was not the principal determination. For calculating the ideal rate, Tables 6 and 7 are used, with the relations and methods of § 16.

TABLE 17. PEABODY EXPERIMENTS

Set.	Length, inches.	p_1	$\frac{p_2}{p_1}$	$\frac{p_0}{p_1}$	W	D_T	W_T	c
A	1.5	88.0	0.33–0.45	0.634	0.0606	69.80	0.0617	0.918
B	0.5	87.5	0.34–0.51	0.616	0.0623	69.80	0.0615	1.013
C	0.25	87.1	0.34–0.45	0.583	0.0627	69.79	0.0613	1.023
D	0.25	140.4	0.30–0.40	0.596	0.0992	70.70	0.0974	1.019

C. H. Peabody, Trans. A.S.M.E., 1890, Vol. 11, 187–192.

In the experiments of Table 17, the steam flowed through what was really a short, straight nozzle: the rounded entrance had a quadrant arc of about 1 in. radius for profile, while the tubular part was 0.25 in.

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in diameter and had the three lengths given in the first column of the table. At mid-length of the cylindrical portion of the nozzle, a small hole was drilled in from the side, and connection was made to a gage which showed the actual "throat" pressure p_0. The four lines in the table are means of groups containing two to five individual tests. The only quantity that had more than incidental variation within the group was the low-side pressure p_2, and this is covered by giving the range of the ratio p_2/p_1; in all cases, p_2 was well below the critical value $0.58\,p_1$. Before considering the calculation of ideal flow, best illustrated by an example, note how well the observed pressure in the tube conforms to the theoretical throat pressure: naturally, it would be higher as the tube was longer and the gage hole farther from the outlet.

EXAMPLE 46. — For line C of Table 17, take from Table 7 the value 69.79 for the Napier divisor D. This corresponds, of course, with the normal or ideal throat pressure $p_0 = 0.58\,p_1$. The area of a 0.25 in. circle is 0.04909 sq. in. Then the theoretical flow per second is

$$W_T = \frac{ap_1}{D_T} = \frac{0.04909 \times 87.1}{69.79} = 0.0613 \text{ lb.} \quad . \quad . \quad . \quad . \quad (237)$$

The measured flow W being 0.0627 lb. per sec., the coefficient of discharge is

$$c = \frac{W}{W_T} = \frac{0.0627}{0.0613} = 1.023. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (238)$$

(*b*) THE RATEAU EXPERIMENTS, represented by Table 18 and plotted in Fig. 326, constitute perhaps the best single body of data on the flow of steam through the convergent nozzle and the plain orifice. Out of about 150 results, selected examples are given in the table, while the nearly 100 points in the diagram show the whole set, since many of these points stand for two or more practically coincident values. The nozzles tested are given in Fig. 325, with orifice diameters and with areas in square inches marked on the drawing. The columns in the table are as follows:

p_1 = initial pressure in pounds per square inch absolute;
R_p = ratio of discharge or low-side pressure p_2 to p_1, or is p_2/p_1;
D_T = theoretical Napier divisor, from Tables 7 and 6, to be used in Eq. (237); see Example 47;
W_T = theoretical flow in pounds per second, by Eq. (237);
W = actual flow, in same terms as W_T;
c = coefficient of discharge, by Eq. (238).

When p_2 does not exceed p_0 or $0.58\,p_1$, D_T is taken directly from Table 7, for the particular value of p_1. If p_2 is above p_0, this D_0 must be enlarged, using the area ratio a/a_0 from Table 6. As a increases,

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TABLE 18. EXAMPLES FROM RATEAU'S EXPERIMENTS ON FLOW OF STEAM

Set.	p_1	R_p	D_T	W_T	W	c
A	151.9	0.890	103.9	0.1559	0.1494	0.959
	143.7	0.738	76.19	0.2008	0.1965	0.979
	151.4	0.466	70.84	0.2280	0.2305	1.011
	138.3	0.015	70.67	0.2083	0.2124	1.020
B	77.1	0.953	149.5	0.1453	0.1370	0.941
	86.2	0.908	110.4	0.2200	0.2100	0.955
	55.7	0.891	101.36	0.1550	0.1474	0.951
	58.3	0.474	68.96	0.2382	0.2417	1.013
	58.8	0.282	68.98	0.2406	0.2446	1.016
	110.7	0.019	70.27	0.4038	0.4130	1.022
C	57.4	0.984	248.0	0.1770	0.1587	0.895
	57.6	0.951	145.9	0.3022	0.2760	0.913
	17.6	0.864	89.36	0.1507	0.1404	0.932
	22.5	0.782	75.92	0.2266	0.2170	0.957
	21.4	0.687	69.54	0.2354	0.2321	0.987
	23.2	0.544	67.02	0.2647	0.2642	0.998
	22.8	0.432	66.98	0.2600	0.2628	1.011
	16.9	0.105	66.32	0.1951	0.1980	1.015
	41.7	0.058	68.25	0.4680	0.4720	1.010
D	70.9	0.965	171.3	0.2048	0.1273	0.622
	46.0	0.838	85.97	0.2650	0.1775	0.670
	41.4	0.640	68.91	0.2977	0.2227	0.749
	58.8	0.396	68.98	0.4218	0.3555	0.843
	54.9	0.257	68.84	0.3943	0.3473	0.881
	57.5	0.039	68.93	0.4130	0.3642	0.883

A. Rateau, Experiments on the Flow of Steam through Nozzles. Published (book form in English) in 1900. Experiments made in 1896.

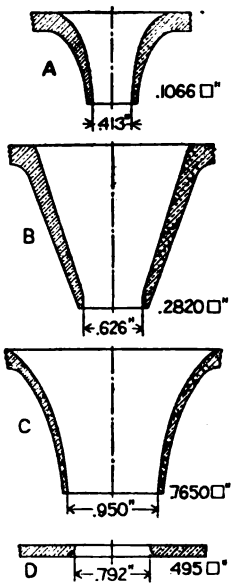

FIG. 325.—Nozzles Used in Rateau's Experiments.

running up the jet from a_0, in exactly the same way must D increase if Eq. (237) is to be used at any cross plane.

The ordinate in Fig. 326 is the coefficient c, on the ratio R_p as base. Curves A, B, C (for the similarly designated nozzles) are separated, each having its own scale, while N is their average to the main scale at the left, which serves for curve D also. Except at the beginning of curve C the results are smooth and consistent, even with irregularities greatly magnified by the coarseness of the vertical scale. Variation in the initial pressure p_1, as distinct from variation in R_p, seems to have very little effect. For the converging nozzle, actual flow conforms to ideal when $p_2 = p_0$, but the coefficient c drops off toward 0.9 as p_2 approaches p_1, and runs above unity when p_2 becomes small. This excess flow with low discharge pressure has not been rationalized, but is amply confirmed by other experiments; further examples, of flow through diverging nozzles, will be found in Art. (*d*). It does not appear

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that the small differences of form in nozzles A, B, and C have any effect upon rate of flow.

With the plain orifice D, flow is relatively smaller, because of contraction of the jet: but the discharge continues to increase quite rapidly

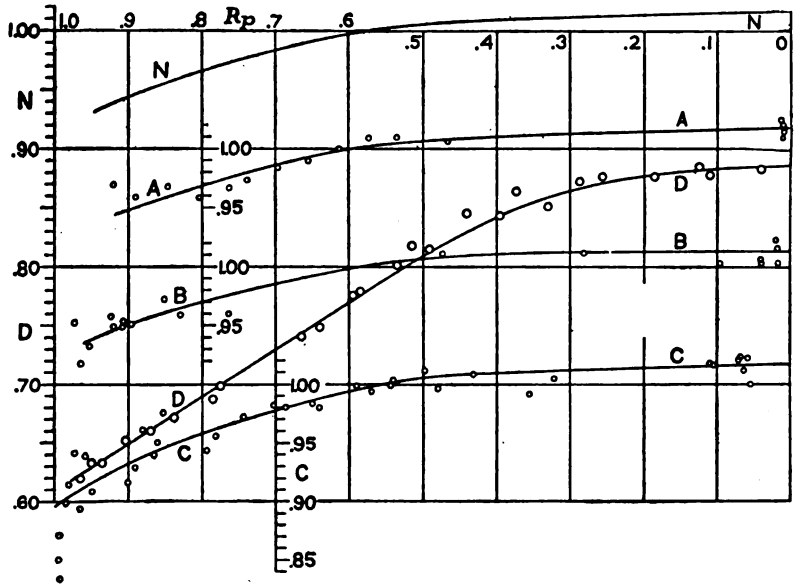

FIG. 326. — The Coefficient of Discharge, as Calculated from Rateau's Experiments.

as p_2 falls below p_0, showing that the influence of contraction is largely dependent upon the density of the medium into which discharge takes place: this action of a thin-plate orifice is, however, in decided contrast with that of tubular orifice A in Fig. 328.

EXAMPLE 47. — In the first line of Table 18, with $p_1 = 151.9$ lb. abs., D_0 from Table 7 is 70.84. With $R_p = 0.890$, the value of a/a_0 from Table 6 is 1.466. Multiplying,

$$D_T = 70.84 \times 1.466 = 103.85.$$

With the orifice area 0.1066 sq. in., the calculated flow is

$$W_T = \frac{0.1066 \times 151.9}{103.9} = 0.1559 \text{ lb. per sec.}$$

(c) FLOW OF SUPERHEATED STEAM. — In Fig. 327 are plotted the results of some large-scale determinations of the flow of superheated steam: the rate was from 8000 to 9000 lb. per hour, the orifice being used to "waste" steam in some special boiler tests. The pressure was 160 lb. abs. for the series given, and efflux was into the atmosphere.

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The diagram is plotted on degrees fahrenheit of superheat as base. The curve marked D_T shows the calculated Napier divisor, worked out for $p_0 = 0.58\ p_1$, as in Example 48, following: it is extended to 300 deg. in the upper right-hand part of the drawing. Corresponding with D_T, W_T is the theoretical flow in pounds per second; and the dotted lines

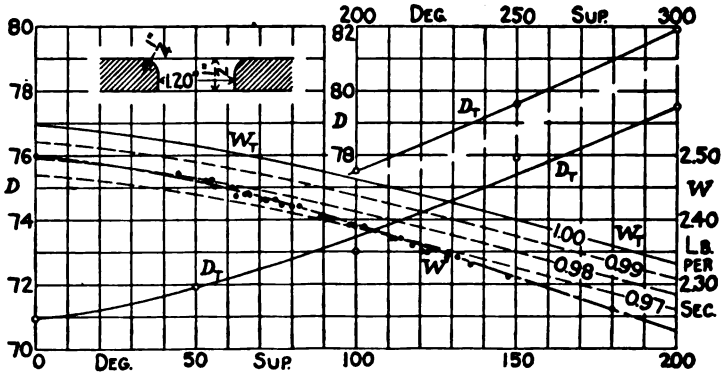

FIG. 327. — Flow of Superheated Steam into Atmosphere, from 160 lb. abs., through 1.2 in. orifice. Tests reported by I. Harter, Jr., Jour. A. S. M. E., Dec., 1910, Vol. 32, 2017–2021.

are at 1, 2, and 3 per cent below the full unit value of this quantity. Experimental flow results having been plotted as W points, coefficients of discharge can be read on the scale furnished by these dotted percentage lines.

At first sight, the fact that this coefficient lies below unity seems to contradict the showing of the last article; but when we note how small is the radius of rounding at entrance to the orifice, a ready explanation suggests itself in the very probable existence of some contraction of the jet. Why the ratio of discharge should decrease with greater superheat is not apparent.

EXAMPLE 48. — Calculate theoretical conditions at 200 deg. of superheat for Fig. 327.

Initial pressure p_1 = ...	160 lb. abs.
Saturation temperature t_{s1} = ...	363.6 deg.
Initial steam temperature t_1 = ...	563.6 "
By Table VIII, at p_1 and t_1, entropy n_1 = . . .	1.6846
Throat pressure $p_0 = 0.58\ p_1$ = ...	92.8 lb.
By Table VIII, at n_1 and p_0, temperature t_0 = . .	439.5 deg.
By Table VII, at p_1 and t_1, total heat h_1 = . . .	1304.5 B.t.u.
By same table, at p_0 and t_0, total heat h_0 = . . .	1248.0 "
Then available energy $E = h_1 - h_0$ = ...	56.5 "

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By Eq. (104), steam velocity $V = 223.7\sqrt{E} = \ldots$ 1679 ft. per sec.
By the method of § 12 (*i*), Example 2,
at p_0 and t_0, specific volume $v_0 = \ldots$ 5.637 cu. ft.
Then area $a_0 = \frac{144\, v_0}{V_0} = \frac{144 \times 5.637}{1679} = \ldots$ 0.483 sq. in.
And $D_T = p_1 a_0 = \ldots$ 77.4
The area of a 1.2 in. circle is $\ldots$ 1.131 sq. in.
Whence the ideal flow per second is

$$W_T = \frac{1.131 \times 160}{77.4} = \ldots \text{ 2.34 lb.}$$

The circle-marked points along the D_T curve in Fig. 327 are from calculations like the preceding. Their erratic departures from a smooth curve are due in part to the impossibility of making very precise readings in the table-diagrams for superheated steam, in part to insufficiently consistent spacing of the curves in these Tables VII and VIII. An error of 0.5 to 0.7 B.t.u. in the value of the available energy E is enough

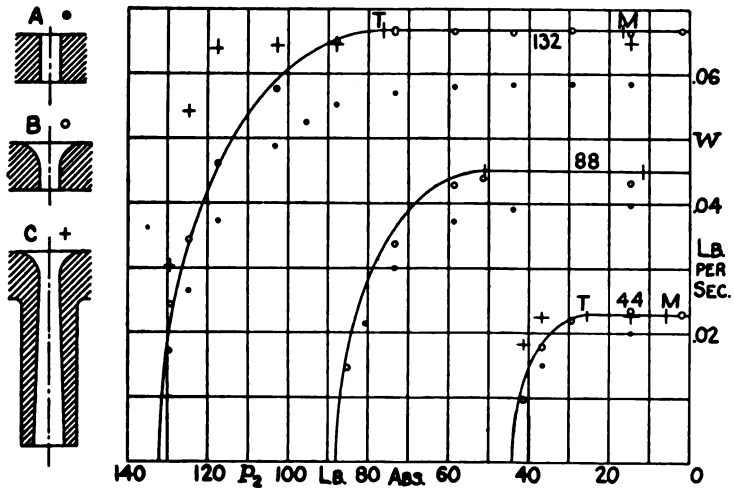

FIG. 328. — Selected Results from Experiments by Gutermuth, *Zeitschrift des Vereines deutscher Ingenieure*, 1904, Vol. 48 I, 75–84. Steam dry-saturated.

to cause the discrepancies which appear at 100 and 150 deg. This calculation is an exceedingly severe test of the accurate interrelation of the diagrammed properties of superheated steam.

(*d*) FLOW THROUGH DIVERGENT NOZZLES. — Experiments made simply to determine the rate of flow are well represented by the results set forth in Fig. 328. The flow pieces are shown at the left of the drawing, all having a least diameter of 5.4 mm. or 0.213 in. Each series

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is for a constant initial pressure, here 3, 6, and 9 atmospheres respectively, or 44.1, 88.2, and 132.3 lb. per sq. in. The discharge pressure p_2 was varied over a wide range, and is the determining abscissa in the diagram. Plotted points show experimental data, while the curves represent calculated flow, with assumed adiabatic expansion, or the regular "ideal" action. On each curve a cross line is drawn at T to mark the throat pressure p_0, another at M to show the muzzle pressure of nozzle C*: the latter has about the area ratio $a_2/a_0 = 2.05$, for which the pressure ratio is $R_p = 0.133$, by Table 6.

Square-shouldered orifice tube A naturally shows a big falling off from ideal flow, but differs from the thin-plate orifice of Rateau, Fig. 326, in that the flow does not continue to increase after p_2 falls below p_0. Orifice B keeps very close to ideal discharge. The most striking thing shown, however, is the way in which points from nozzle C lie far above the calculated curve for values of p_2 higher than p_0, so that the nozzle reaches full discharge with p_2 as great as $0.8\,p_1$. These experiments

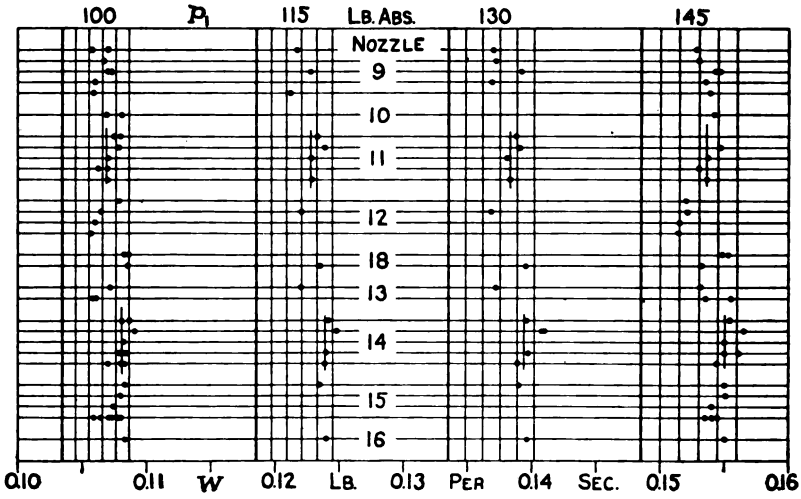

FIG. 329. — Rates of Flow, Tests of Steam-turbine Nozzles, Sibley and Kemble, Trans. A. S. M. E., 1909, Vol. 31, 617–653. Nozzles in Fig. 330, discharge into vacuum.

have been cited more to exhibit this phenomenon than because of any especial quantitative value which they possess for present purposes: but before the matter of overexpansion in the nozzle is taken up, the further flow tests in Fig. 329 will be considered.

In these Sibley and Kemble tests, which deserve decided prominence

* That is, the pressure to which the steam will be expanded when the ideal jet has the same degree of divergence as that of this nozzle.

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and will be more fully discussed presently, the four initial pressures marked along the top of Fig. 329 were used, and the points grouped beneath each value represent tests at (or from) that pressure. The abscissa of the diagram is, of course, the rate of flow, in pounds per second to the scale at the bottom. At the left of each group, a heavy vertical line shows the calculated ideal flow, with steam initially dry-saturated and with an orifice area of 0.0725 sq. in. — all measured rates being reduced to this prevailing area. To the right of each ideal line are drawn five percentage lines, at intervals representing one per cent of the ideal flow; then coefficients of discharge can be read off directly. The short vertical lines across sets 11 and 14 will be referred to from Art. (*k*). The vertical spacing has no quantitative meaning, but each horizontal line ties together the tests made on a certain date. For the present we note simply that the coefficient of discharge ranges from 1.02 to 1.05.

In Fig. 330, beside the length and angle of divergence, each nozzle profile carries three numbers; these are: beneath the throat, the throat

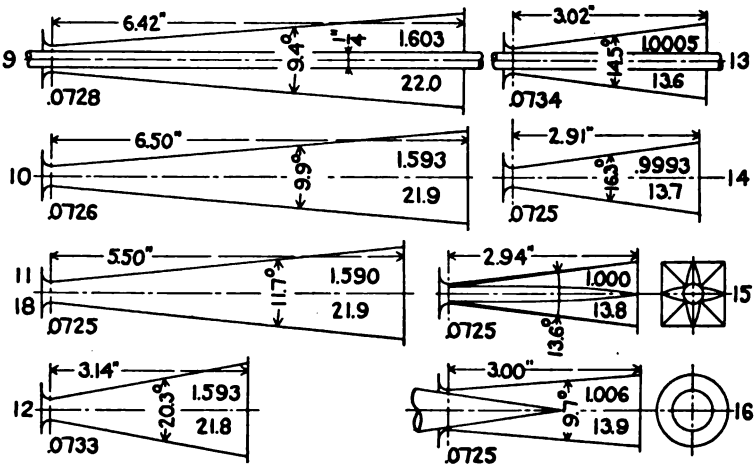

FIG. 330. — Nozzles tested by Sibley and Kemble.

area a_0 in square inches; near the muzzle, first the mouth area a_2, then the area ratio a_2/a_0. All the nozzles were of machinery steel, bored smooth and polished; except that No. 18, otherwise like 11, was left with a fine tool mark, which, however, does not appear to have had any effect upon rate of flow. Nozzle 15 was round in approach and throat, but was made in halves and milled out to a square muzzle. Nos. 9 and 13 are search-tube nozzles, and No. 12 has an unusually wide cone angle.

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(*e*) MEASUREMENTS OF PRESSURE IN THE JET. — From the beginning of definite investigation of the steam jet, the variant pressure within the steam current has been one of the important quantities measured; and a search tube lying in the axis of the nozzle as outlined in Fig. 331, is the best and most used device for this purpose. One or more holes are drilled through the tube at a particular point in its length: one end is closed, the other goes out through a stuffing box and is connected to a pressure gage. The tube can be moved lengthwise, and an external measuring arrangement determines the position of the holes along the nozzle axis, usually referred to the cross plane of the throat as datum position.

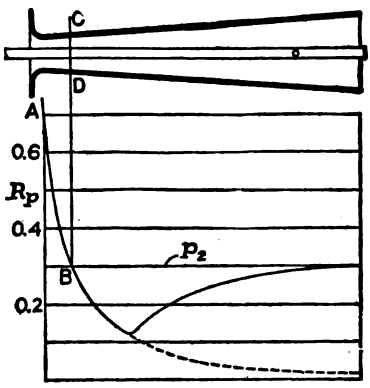

FIG. 331. — Nozzle with Search Tube; overexpansion shown.

Paralleling the scheme of the search tube, we have that of drilling holes through the sides of the nozzle, already described, in limited application, in connection with Table 17, Art. (*a*). This method lacks flexibility in change of locus, and calls for complex gage connections. With a straight-line profile (in a cylindrical or conical tube), so that no curvature of path evokes transverse inertia of the stream elements, only an accidental condition will cause any difference between pressure measurements at the axis and at the envelope of the jet; so long as orderly flow is maintained, it is immaterial where the pressure is taken, in a given cross plane.

In Stodola's *Steam Turbines* will be found an account of some comparative pressure measurements, made with holes in the sides of search tubes drilled obliquely (at 45° slant), some inclined against, some with the current. The observed pressures differed in very uniform fashion, and their mean was just about equal to the reading with holes drilled perpendicular to the surface. That the latter arrangement gives essentially correct results is the conclusion announced.

(*f*) CURVES OF PRESSURE VARIATION. — The diagram below the nozzle in Fig. 331 shows first, in the curve AB (extended in dotted line), the characteristic pressure-distance curve for the conical nozzle. Note how very rapidly the pressure falls just beyond but near to the throat, or how little enlargement of section is needed to accommodate a big range of pressure drop in that region.

The same diagram is further intended to illustrate what happens when the nozzle expands too much, or has too great an area ratio for

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the existing pressure ratio. In the figure, with R_p equal to 0.3, drop to this p_2 is accomplished at CD, and the nozzle should end there. In the extension of the nozzle, orderly jet formation continues to some lower pressure — we may say, of its own inertia: then a sudden change of condition and action occurs, and the pressure begins to rise. This

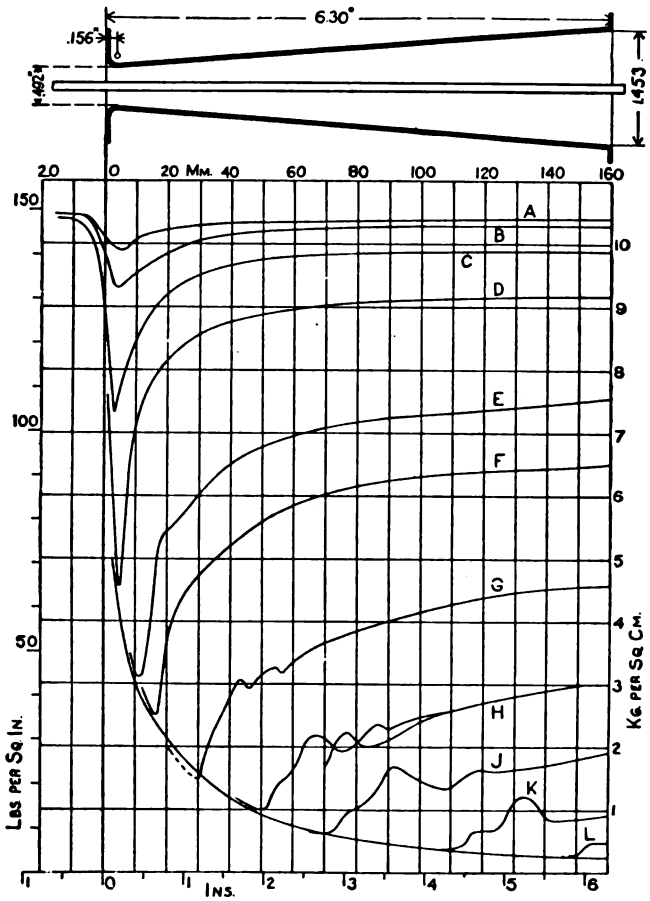

FIG. 332. — Curves of Pressure Variation in a Long Nozzle: experiments by Stodola, diagram redrawn from Stodola's *Steam Turbines*.

involves a considerable dissipation of progressive velocity into secondary motions, and the reconversion of kinetic energy into pressure work as the stream is retarded and compressed. By the time the jet gets to the mouth of the nozzle, it has far less available energy than at B, and a correspondingly larger heat content.

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The induction of a local pressure much less than the final p_2 accounts for the early establishment of full flow in the divergent nozzle of Fig. 328. In effect, the nozzle exerts a sort of suction; and when p_2 is greater than p_0 this induces at the minimum section a pressure lower than p_2, hence gives increased flow. But as soon as the zone of least pressure passes beyond (to the right of) the throat, the flow becomes practically fixed in rate.

Figure 332 serves as a very good representative example of results got by means of the search tube, covering a wide variation of final pressure with a nearly constant initial pressure. This diagram shows the realized form of the action which is merely sketched in Fig. 331. The rise of pressure just beyond the minimum is almost an abrupt jump, indicating something like a shock in the change of flow condition. At the lower discharge pressures, curves G to K, appear the wave lines of incipient acoustic vibration. The sound wave, having a fore-and-aft oscillation in the direction of its propagation, sets up in the steam current alternating zones of higher and lower pressure.

Stodola, to whom the reader is referred for further information, gives a number of sets of curves of this type. Many of them come from what obviously are not proper forms for steam-turbine nozzles, and, by the exceeding badness of the action which they show, fulfill the intention of confirming the conclusions of reason and common sense as to what are likely to be proper forms. Of greatest interest after Fig. 332 is a group of curves from a convergent nozzle, or "orifice" with rounded entrance: the investigation is carried well beyond the plane of outlet, and shows a very rapid drop of pressure when p_2 is less than p_0. Well-marked vibrations appear, of moderate magnitude when p_2 is about equal to p_0, increasing as p_2 falls, then diminishing as p_2 becomes very small.

(*g*) TEMPERATURE GRADIENT IN THE JET. — Complementary to the idea of localized pressure measurement, that of similar temperature measurement has been applied in a few experiments. Sample results are given in Fig. 333, which shows observations made by both methods, in the De Laval nozzle at the top of the figure: steam of 60 lb. initial gage pressure was discharged into the atmosphere, this initial pressure being so low that there was overexpansion in the nozzle. The curves or point series, taken in order as lettered, are as follows:

Curve A, to scale at left, is pressure measured by means of a search tube which ran clear through the nozzle and had a hole drilled squarely across it, in the usual fashion.

Curve B is pressure measured by a search tube open at the end and projecting into the nozzle from the entrance side, so that the opening

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was in the direction of flow. It is evident that the abrupt enlargement of cross area at the squared-off end of the tube must have had a very considerable effect upon the local pressure gradient, and that the difference between B and A is by no means all due to suction of the eddy at the end of the tube. In fact, beyond the beginning of the sloping nozzle mouth, where area ceases to be determinate, the two pressures become scarcely distinguishable.

Curve C shows measurement of pressure by an open-end tube thrust

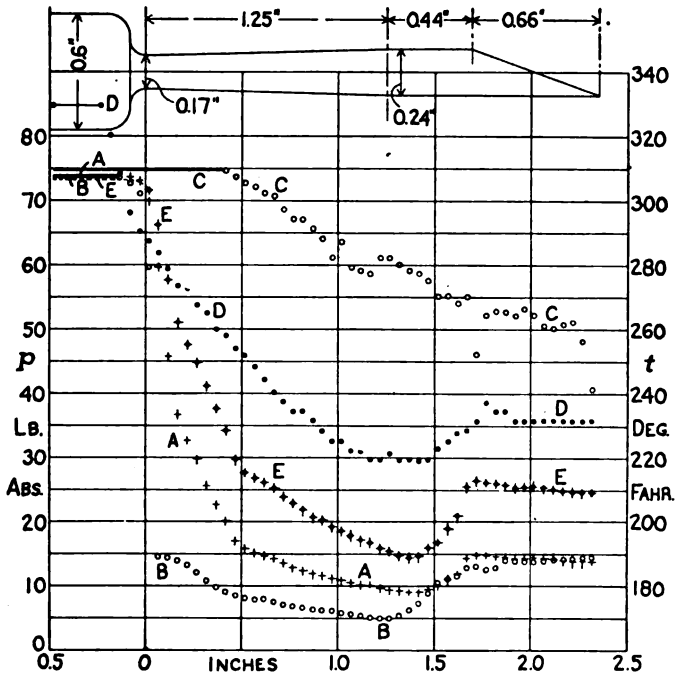

FIG. 333. — Results of Pressure and Temperature Measurements, tests by Borsody and Cairncross at Columbia University. Reported by Prof. C. E. Lucke, Trans. A. S. M. E., 1905, Vol. 26, 114–158.

into the nozzle from the discharge side. Here again, abrupt change of section must produce some disturbance of flow; but it appears that a large part of the velocity "head" is converted into static pressure in the measuring tube, even when the density of the steam in the jet has become quite low.

Curve D, to scale at right, gives the temperature in the steam, as measured by a thermocouple, of which the joined wire was stretched along the axis of the nozzle and moved lengthwise just like the pressure tube.

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Curve E is the temperature of saturated steam corresponding with measured pressure A, laid out in order that D may be compared with it. The steam was superheated at the start. In very peculiar fashion, temperature D at first falls below E; but it soon reverses relation and runs from 30 to 20 deg. above E. Rationally, after nearly adiabatic expansion has (very soon) brought the steam out of the region of superheat into that of saturation, the temperatures ought to coincide. If curves D and E were fairly near to each other, and a close comparison

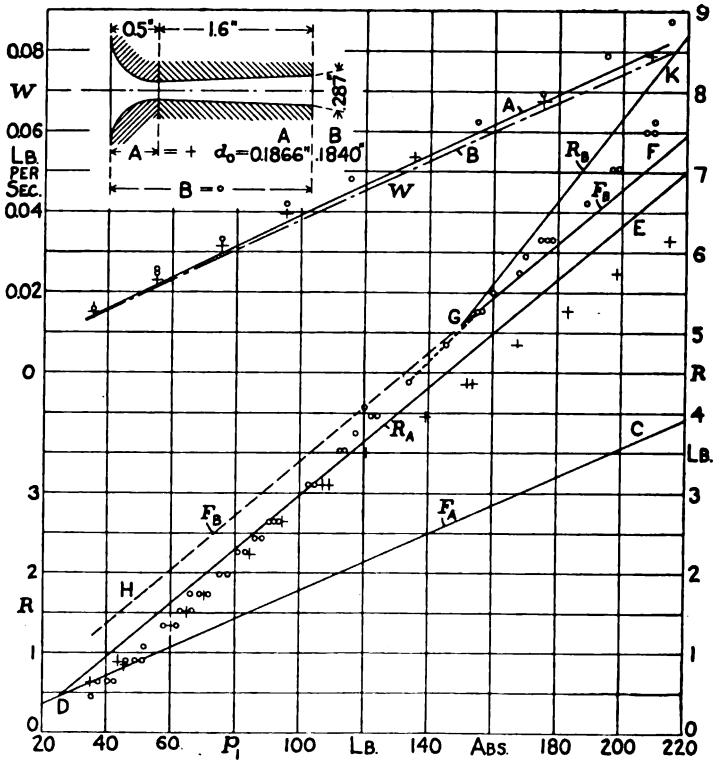

FIG. 334. — Reaction upon the Nozzle, tests by W. Rosenhain, at Cambridge University. Proc. Inst. C. E., 1900, Vol. 140, 199–220.

worth while, difference in effective variant area ratio and in pressure gradient, due to the substitution of small wire for larger tube, would have to be taken into account in bringing them to a common basis. With actual data, the comparison has nothing but the roughest qualitative value.

In connection with the preceding may be cited some experiments by C. Batho, at the University of Liverpool, reported in Proc. Inst.

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C. E., 1907–08, Vol. 174, 317–331. A thermocouple was used in essentially the same way. With a brass steam nozzle, the observed temperature ran a little below that from the calculated pressure gradient until near the end of expansion, then rose above it. With a porcelain nozzle, lined with glass which was carefully ground to form and polished, the measured temperature kept about 20 deg. above the theoretical. Some rather unwarranted conclusions were drawn as to relative losses of heat or transfer along the nozzle.

The matter seems to be best summed up by the statement that the temperature method has shown itself distinctly unreliable in this particular line of work.

(*h*) Efficiency in Jet Formation. — This is measured by the ratio of actual kinetic energy of the jet, as delivered by the nozzle, to calculated available energy of the steam according to the Rankine cycle. Actual energy becomes known if we can find the velocity of the steam: and while velocity itself would be exceedingly difficult to measure, the simple relation of velocity to impulse or reaction leads at once to the idea of measuring that force.

Experiments have been made in which a flat plate was held in front of a nozzle, at a short distance from the mouth,* as indicated in Fig. 303. These give a fairly good determination of impulse, but it is difficult to eliminate secondary disturbances which interfere with accuracy. The scheme outlined in Fig. 304 is inherently rather better, but must be used with due regard to the considerations set forth in § 46 (*b*) and Examples 42 and 43. We shall first review some earlier experiments in which the relations involved were not fully taken into account. These will lead up to some thoroughly correct tests, which give very good illustration of the reaction method, and also of the indirect method of calculating efficiency from observed pressure in the jet.

(*i*) Rosenhain's Reaction Tests. — In Fig. 334, the principal series of plotted points shows measured reactions, due to discharge into the atmosphere of steam of varying initial pressure, through the rounded orifice A and the full nozzle B as outlined at the top of the diagram. The points along the lines marked W give rates of flow in pounds per second, two lines of calculated flow being necessary because of slightly different minimum diameters in the two cases. For comparison with the reaction points, a number of theoretical curves or lines are drawn, as follows:

Line CD shows the dynamic reaction F_A resulting from expansion to $p_0 = 0.58\, p_1$ in orifice A, calculated after the manner of Example 42,

* For example, by E. Lewicki at Dresden, published in *Zeitschrift des Vereines deutscher Ingenieure*, 1903, page 491.

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using the ideal discharge in Eq. (237). Based on Tables 6 and 7, which assume similarity of jets from different initial pressures, when referred to the pressure ratio R_p, CD is a straight line.

Line ED adds to F_A the static-pressure reaction $(p_0 - p_2) \times a_0$, giving the total calculated reaction R_A. At D, where $p_1 = 14.7 \div 0.58 = 25.4$ lb. abs., the orifice pressure p_0 is just equal to the external pressure, and the reaction measured is purely dynamic. Above D, or for practically the whole range of this particular experiment, the observed reaction is of no value at all as a criterion of jet formation.

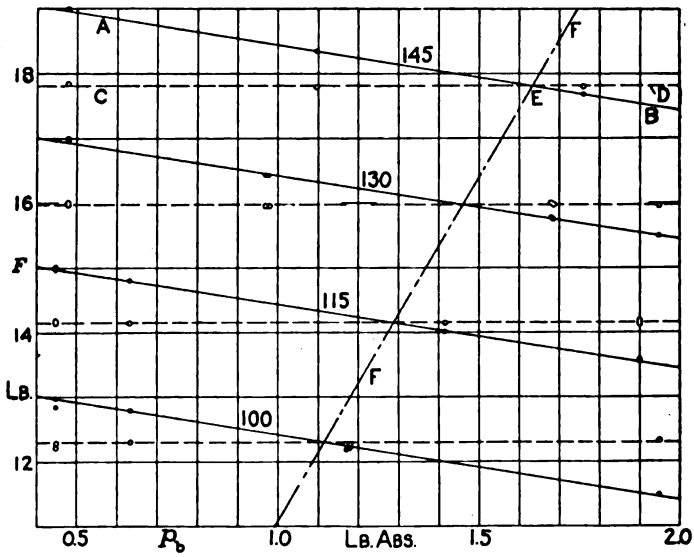

FIG. 335. — Observed and Corrected Reactions, Sibley and Kemble Tests. See reference under Fig. 329. These data belong to nozzle 14 in Fig. 330.

With the expanding nozzle B, lines FG and KG give F_B and R_B in the same manner. These lines meet at a comparatively high pressure, since the nozzle has an area ratio $a_2/a_0 = 2.435$, for which $R_p = 0.098$ by Table 6: wherefore the initial pressure which will just reduce to that of the atmosphere is $p_1 = 14.7 \div 0.098 = 150$ lb. abs. With p_1 less than 150, there is overexpansion in the nozzle. If, as is probable, the variant pressure p rises to p_2 within the nozzle, as indicated in Fig. 331, comparison of measured R_B with the ordinate of line GH will show the proportional loss of velocity caused by excessive divergence.

Only at or near G, where a short curve is traced through several experimental R_B points, can the efficiency of the nozzle properly be derived from these experiments. At G, the measured reaction is only

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1.6 per cent below the ideal; but as appears from the W comparison, the former is produced by about 1.05 times as much steam as the latter. Dividing 0.984 by 1.05, we have 0.933 as the ratio of reactions, or of actual to ideal velocity. Squaring this ratio, because kinetic energy varies as square of velocity, we get an efficiency ratio of 0.87.

It is evident that any comparison among different nozzles in this group of tests, based simply on observed reactions, without any analysis such as has just been exemplified, must be inconclusive and of little value.

(*j*) THE SIBLEY AND KEMBLE TESTS. — These experiments, already introduced in Art. (*d*), combine the reaction and the search-tube method in a manner which is really necessary if definite results are to be obtained by the former. The arrangement was essentially that of Fig. 304, and similar to Rosenhain's except that both the nozzle chamber and the flexible steel tube carrying it were enclosed in a box which was connected to a condenser. The dynamometer was a pulling spring, and with any load this was adjusted so as to bring the reaction chamber to its neutral position, thus eliminating any elastic resistance of the tube to bending: the extension of the spring, or the displacement of the screw slide carrying its front end, measured the reaction. For search-tube work, the nozzle box was clamped, and the tube coupled to a gage connection. Only the pressure in the mouth of the nozzle, at or just within the terminal cross plane, was sought: and beside the search tube in nozzles 9 and 13, Fig. 330, every nozzle had a small test hole drilled in its wall just inside the end plane.

The purpose in measuring the mouth or muzzle pressure in the nozzle will be made clear by a study of Fig. 335, which shows a typical set of reaction observations, one group for each of the four initial pressures already named in Fig. 329. The base is absolute pressure p_b in the vacuum box surrounding the nozzle chamber, called the "box pressure." With a fixed initial pressure, as $p_1 = 145$ lb., and with a resulting constant terminal pressure p_2 due to expansion within the nozzle to the final cross area a_2, the measured reaction varies as shown by the points along the inclined line AB: it is greater with a low box pressure, less as p_b increases — this being in accord with the statements in § 46 (*b*), under Fig. 304. Subtracting or adding the force got as the product of pressure difference $(p_2 - p_b)$ by area a_2, the points along the horizontal line CD are found. The height of this line is the true dynamic reaction, as the mean of a number of determinations; and the intersection E shows, on the base scale, the terminal pressure p_2 in the nozzle. A simple locus EF, within the limits of these experiments a straight line, connects the several values of p_2.

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TABLE 19. EFFICIENCY RESULTS, NOZZLE TESTS BY SIBLEY AND KEMBLE. NOZZLES NOS. 11 AND 14 IN FIG. 330.

Quantity.	Nozzle 14.				Nozzle 11.			
p_1	145	130	115	100	145	130	115	100
p_2	1.632	1.460	1.288	1.116	0.929	0.832	0.735	0.638
F	17.82	15.98	14.15	12.30	18.13	16.24	14.35	12.45
W	0.1550	0.1394	0.1239	0.1081	0.1536	0.1383	0.1228	0.1069
V_r	3698	3685	3672	3659	3796	3776	3759	3744
E_r	273.2	271.3	269.5	267.5	288.0	285.0	282.4	280.1
E_t	290.9	289.5	287.8	286.1	318.6	317.1	315.3	313.2
e_r	0.939	0.938	0.937	0.935	0.904	0.899	0.895	0.895
a_2	6.45	7.16	8.06	9.24	10.37	11.52	12.98	14.92
a_t 1.00	6.32	7.05	7.96	9.16	10.04	11.18	12.65	14.54
a_t 0.95	6.59	7.36	8.32	9.56	10.51	11.70	13.23	15.21
a_t 0.90	6.90	7.69	8.72	10.00	11.00	12.26	13.85	15.93
e_a	0.976	0.982	0.986	0.990	0.965	0.967	0.972	0.975

Experiments by F. H. Sibley and T. S. Kemble, at Case School of Applied Science, Cleveland, Ohio. Trans. A. S. M. E., 1909, Vol. 31, 617-653. All quantities below E_r recomputed or newly computed by the writer, with values from the steam tables in this book.

(*k*) CALCULATION OF EFFICIENCY. — Having initial pressure and quality of steam, rate of flow, reaction on the nozzle, and terminal pressure within the nozzle, efficiency in the generation of kinetic energy may be calculated in two ways, both of which are illustrated in Table 19. The description of this table may well be accompanied by an outline of the principal calculations, with numerical values from and for the first column. The various quantities and their derivation are as follows:

p_1 initial steam pressure, pounds per square inch absolute. At p_1 the steam is taken to be dry-saturated: in some tests it showed a trace of superheat, in others there may have been a little moisture present.

p_2 is the terminal nozzle pressure, determined as in Fig. 335. Note the lower values prevailing with the larger area ratio of nozzle 11.

F corrected dynamic reaction, also determined by the construction in Fig. 335.

W rate of flow or discharge, in pounds per second. On Fig. 229, short vertical lines in the test groups for nozzles 11 and 14 show the flow values used by the experimenters.

V_r velocity by reaction, from Eq. (214);

$$V = \frac{Fg}{W} = \frac{17.82 \times 32.16}{0.1550} = 3698 \text{ ft. per sec.}$$

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E_r kinetic energy of the jet per pound of steam, reduced to heat units;

$$E = \frac{V^2}{2g \times 778} = \frac{13{,}675{,}200}{64.32 \times 778} = 273.2 \text{ B.t.u.}$$

E_t available energy per pound of steam, within the limits p_1 and p_2, by the methods of § 15 (*d*): detailed calculation in Example 49, following.

e_r efficiency as found by the reaction method;

$$e_r = \frac{E_r}{E_t} = \frac{273.2}{290.9} = 0.939.$$

a_2 terminal area of a unit jet similar to that actually observed. The muzzle area of nozzle 14 being 0.9993 sq. in. (Fig. 330) and the rate of flow 0.1550 lb. per sec., the unit area is

$$a_2 = \frac{0.9993}{0.1550} = 6.45 \text{ sq. in.}$$

a_t calculated area a_2, with various efficiencies. Value marked 1.00 is for the ideal jet with adiabatic expansion; those marked 0.95 and 0.90 are for these efficiencies. That is, we assume that 5 or 10 per cent of the available energy E_t is retained in or returned to the steam as heat. This makes velocity V less and specific volume v greater, hence increases a when calculated by Eq. (105): see Example 50, on page 471.

e_a efficiency by the search-tube method. Interpolating a_2 between the 0.95 and the 1.00 values of a_t, we have that 6.45 is 14/27 of the way from 6.59 to 6.32, so that the efficiency must be 0.52 of the way from 0.95 to 1.00, or at 0.976.

EXAMPLE 49.— The calculation of the ideally available energy E_t for the first column of Table 19 is as follows:

Initial pressure p_1 = ...	145 lb. abs.
Initial quality x_1 = ...	1.00
Final pressure p_2 = ...	1.632 lb. abs.
Final temperature, Table II, t_2 = ...	118.78 deg.
Final total entropy at P_2, N_2 = ...	1.93482
Initial entropy, at p_1, $N_1 = n_1$ = ...	1.57248
The value of n_1 remaining constant during adiabatic expansion, the steam at p_2 has from full vaporization the shortage $N_2 - n_1$ = ...	0.36234
At p_2, the entropy of vaporization is b_2 = ...	1.7725
Then the fraction of condensation is $m_2 = (N_2 - n_1) \div b_2$ = ...	0.20443
Latent heat at p_2, r_2 = ...	1025.2 B.t.u.

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Total heat at p_2, H_2 = ...	1111.81 B.t.u.
Heat deficiency, $m_2 r_2$ = ...	209.57 "
Actual heat content at p_2 and m_2, h_2 = ...	902.24 "
Initial total heat H_1 or h_1 = ...	1193.17 "
Available energy $E = h_1 - h_2$ = ...	290.93 "

(*l*) EFFICIENCY BY CONDITION OF JET. — The scheme of the second derivation of efficiency in Table 19, leading to e_a, is illustrated in Fig. 336. For various ratios of realized kinetic energy to quantity of heat ideally convertible, are worked out and plotted, first, the area a of the unit or

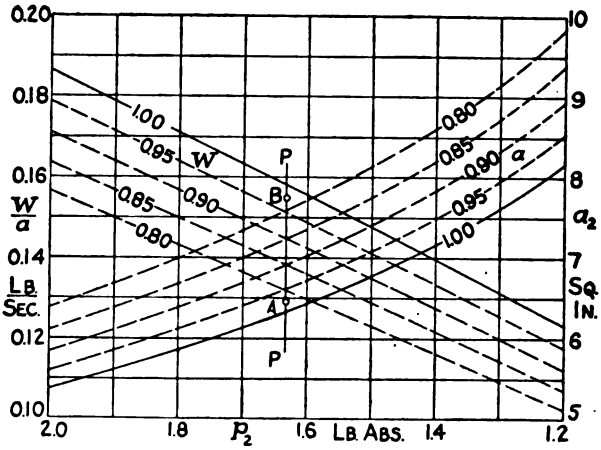

FIG. 336. — Calculated Terminal Proportions of the Steam Jet. Initial pressure uniform at 145 lb. per sq. in. absolute, initial quality 1.00, terminal pressure variable.

pound-per-second jet in square inches; second, the rate of flow W/a in pounds per square inch per second. Line PP is drawn at the terminal pressure $p_2 = 1.632$ of column 1 in Table 19. Efficiency e_a can be found either by interpolating point A between the a-curves for 1.00 and 0.95, as above, or by interpolating W/a at B. In order to make this determination, the rate of flow and the pressure in the steam at an accurately located section of the nozzle must be known. With more emphasis placed upon a certain area than upon a local pressure, point A would be interpolated on a horizontal line, for the observed p_2; but what is here vertical interpolation has the advantage that the area corresponding to a given pressure and efficiency can be calculated, while the theoretical pressure at a given area cannot, but must be found indirectly, from a graphical layout of results by the other method.

In the procedure just outlined, nothing is said about "ideal" discharge, as calculated from the throat area, so that relations are simpler

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than in Fig. 334. For the ideal case of unit efficiency, identity with that rate exists; but in all other cases it is better, as here, to confine attention to the particular cross section under consideration.

EXAMPLE 50. — Find the three a_t values in the first column of Table 19, using quantities from the last example as convenient.

The full ideal energy is E_t = ... 290.9 B.t.u.
One-tenth of this is ΔE = ... 29.1 "
With r_2 = 1025.2, addition of 29.1 B.t.u. to the heat content will change the moisture fraction by the amount $\Delta m = 29.1 \div 1025.2$ = ... 0.0284
Then for the three cases of 1.00, 0.95, and 0.90 efficiency, m will have the respective values 0.2044, 0.1760, 0.1476
The velocity with E = 290.9 is by Eq. (104),
$V_{1.00} = 223.7 \sqrt{290.9}$ = ... 3815.7 ft. per sec.
For the other cases, instead of repeating this calculation, it is easier to look up the square roots of 0.95 and 0.90, getting 0.9747 and 0.9487;* then subtract from $V_{1.00}$ the fractions 0.0253 and 0.0513 of itself, with the results,
$V_{0.95}$ = 3719.2, $V_{0.90}$ = ... 3620.0 ft. per sec.
The full specific volume at p_2 is s_2 = ... 210.3 cu. ft.
For the 1.00 case, $m_2 s_2 = 210.3 \times 0.2044$ = . . . 43.0 "
Also, for 10 per cent change, $\Delta m \times s_2$ = ... 5.96 "
Then the respective volumes are
$v_{1.00} = 210.3 - 43.0$ = . 167.3 "
$v_{0.95} = 167.3 + 3.0$ = . 170.3 "
$v_{0.90} = 167.3 + 6.0$ = . 173.3 "

Now by Eq. (105),

$$a = \frac{144\,v}{V}, \quad a_{1.00} = \frac{144 \times 167.3}{3816} = \ldots \quad 6.32 \text{ sq. in.},$$

and the other areas are found in the same way.

(*m*) RESULTS AS TO EFFICIENCY. — Considering first the results in Table 19, we note in e_r a prevailing difference of nearly 4 per cent between nozzles 14 and 11. With the longer expansion in No. 11, there must be a little more energy lost; but it does not seem reasonable that the ratio of this loss to the available energy (itself a rapidly increasing quantity) should increase in any marked degree. To use equal flow rates, instead of one per cent less for No. 11 as appears in Fig. 329, would bring the efficiencies nearer together — remembering that a small proportion of change in velocity V is doubled when squaring V to get energy E.

* The idea being to use the slide rule for as much as possible of the numerical work.

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Why the efficiency e_a by area should be so much larger than e_r by reaction is not apparent; but it seems that more credence ought to be given to the results by measurement of dynamic force than to those found by a very indirect derivation. The experiments set forth in Table 19 are, so far as the writer knows, the best that have been made (and published) to determine the efficiency of the operation of forming a steam jet in the divergent nozzle. They illustrate fully the need of great accuracy in the measurement of the principal quantities; and the marked difference between results by the two methods, or between e_r and e_a, indicates most decidedly that there is much yet to be learned about the intimate detail of the behavior of the steam in the jet. The problem is, however, one for the physicist rather than the engineer.

As regards the various nozzles shown in Fig. 330, reactions were measured with Nos. 10, 12, and 15, beside 11 and 14. No difference seems to have resulted from the wide angle of No. 12 or the change of shape in No. 15.

In Art. (*i*), from experiments perhaps not very exact in rate of flow, we deduced an efficiency of 0.87 for a De Laval nozzle discharging into the atmosphere. Stodola, in Fig. 27 of second edition of *Steam Turbines*, shows results by the method of Art. (*l*) from a nozzle about 0.5 in. in throat diameter and discharging 0.34 lb. of steam per sec. from 149 lb. abs. into a good vacuum: the plotted points agree very closely with the curve for 10 per cent loss of energy. Continuing, he quotes determinations of Delaporte and of Lewicki by the impulse method suggested in Fig. 303, which gave efficiencies of 0.95 and 0.93, with flow from De Laval nozzles into the atmosphere.

Briling,* using this same impulse measurement with cylindrical and with slightly convergent nozzles, and discharging into atmosphere from initial pressures so low that the ratio $R_p = p_2/p_1$ does not fall below 0.58, gets a ratio of from 0.92 to 0.97 for actual to ideal velocity, or an efficiency of 0.85 to 0.94 — this increasing with higher velocity or greater pressure range.

Experimental knowledge on this subject may be summed up in the statement that the probable efficiency of properly proportioned nozzles is from 0.88 to 0.93. By proportion is meant especially the ratio of divergence or the area ratio a_2/a_0. In general, it is decidedly more harmful to have this too large, causing overexpansion, than to have it too small. As appears from Fig. 336, if the area a_2 be designed for the ideal case it will be a little small for actual conditions; wherefore the ideal proportions may properly be used with little or no modification.

* *Zeit. Ver. deutsch. Ing.*, 1910, Vol. 54 I, 265, etc. Verluste in den Schaufeln von Freistrahldampfturbinen.

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(*n*) FLOW IN CURVED CHANNELS. — As may be inferred from the calculation of centrifugal pressure in Example 44, page 438, a current of steam flowing in a curved channel is subject to forces which certainly do not tend to maintain orderly flow in "parallel" stream lines, on the hypothesis represented in Fig. 305. With any considerable width of channel relative to radius of curvature, such as always exists, there must at the very least be a crowding of the stream against the outer wall, after the manner depicted at I in Fig. 337. But it is evident that this sketch gives a very inadequate idea of what takes place: rather, with a stream entering tangentially as at II in Fig. 337, the curved guide surface may be regarded as cutting obliquely (and quite sharply) across the line of flow; then this surface will not only deflect the current (or give it transverse acceleration) but will also produce some linear retardation. Instead of sweeping smoothly around the curve, the jet will pile into the bucket in a confused and tumultuous fashion, with a rise of average pressure throughout the confined space; and from this broken and compressed stream the escaping jet will again be formed, with pressure drop, in a manner much like that which of intention prevails in the reaction turbine.

Observations of Pressure. — Stodola (Fourth Edition, Berlin, 1910, pages 95 to 102) gives the results of experiments made to find the variant pressure within curved nozzles and blade channels. These were of rectangular cross section, and one plane wall was a movable plate, with a search hole which could be brought to any desired position. The curved nozzles were of the general form of those in Fig. 19, or much like the ordinary reaction-turbine blade channel: there was some "banking up" of the current against the concave side, but comparatively little, because by the time high velocity was attained the channel had become nearly straight. With a row of impulse vanes held in front of the nozzle, and with observations represented by curves of equal pressure very similar to the curves of equal altitude on a topographical map, some interesting results are shown, in Stodola's Fig. 88. The nominal expansion was from 155 lb. absolute to atmospheric pressure; but the standing vanes backed the pressure up to about 28 lb. in the mouth of the nozzle. At the middle cross section of the blade channel which, being squarely in front of the nozzle, received full inflow — the plane of this section being perpendicular to the channel walls and to what would be the axis of wheel rotation — the pressure varied from about 105 lb. against the outer guide surface, or in the bottom of the bucket, to about 40 lb. near the inner surface; and at the outlet of the channel it was still about 20 lb. abs.

Increase of Radial Depth. — Of course, this action involves a loss of

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progressive kinetic energy and of stream velocity. Its most striking practical effect is seen in the prevailing departure of construction from the conclusions stated at the end of § 46 (*k*). With no secondary losses, no increase in the radial length of the vane, or in the depth of channel, would be called for; but because of large bucket loss such as has just been indicated, experience has shown the need of a decided increase of radial depth, to provide a sufficient enlargement of flow area — see the sections of Curtis wheels in Figs. 23 and 359. Without such gradual enlargement, the current would back up and the pressure be higher on the nozzle side of the wheel. These remarks refer especially to multiple-impulse wheels of the Curtis type, but apply in a lower degree to single vane rows in the stage.

Non-symmetrical Vanes. — If a vane channel is curved clear to exit, as in Fig. 337 II, the stream will have some tendency to curl over the outer edge, swinging away from the tangential direction there

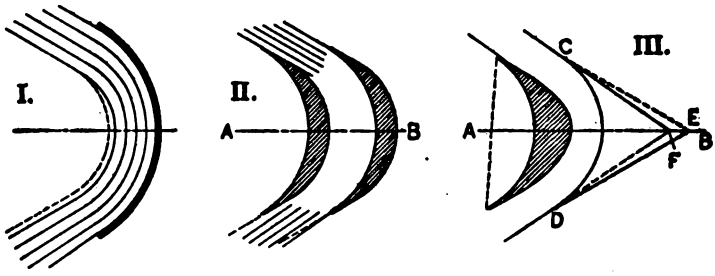

FIG. 337. — Behavior of the Jet in the Bucket or Vane Channel.

represented by dotted lines. To compensate for this and get a larger driving component from the reaction of the escaping jet, impulse vanes are now often made as shown at III in Fig. 337, with a narrower angle of exit ABD than of entrance AFC.

Limitation of Velocity Staging. — The obvious impossibility of making a steam current of intended uniform pressure follow a tortuous path in orderly flow accounts for the limitation which experience has placed upon velocity staging, or upon what in this book is called multiple-impulse action. Thus in the service of driving electric generators, large Curtis turbines are now never made with more than two vane rows to the wheel, as against three in a few early examples. The use of three or in some cases four rows on a wheel, under the lower-speed requirements of marine service, must involve loss of efficiency. The Riedler-Stumpf type of arrangement, with tangential buckets and with long curved return channels, as shown in Fig. 354, has been definitely given up for large and efficient turbines.

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In this connection, it is of interest to refer to some results given by Stodola (Fourth Edition, page 100) of pressure measurements along the guide channels of a small Elektra turbine. The machine was generally similar to that in Fig. 352, but had only one pressure stage, with four impulses. At the four places where the steam current, after leaving the nozzle, flowed between and acted upon the moving vanes, it had the pressures 4.7, 6.4, 2.8, and 1.5 lb. abs., a considerable rise occurring in the first guide channel. A descending pressure gradient beyond the nozzle is, of itself, an influence toward better performance; and this particular turbine is better adapted than are most small multiple-impulse designs to the task of holding in a definite channel a current of varying pressure, without much leakage or spilling.

(*o*) IMPULSE UPON VANES. — A good many experiments have been made in which a group of turbine buckets was held in front of the nozzle in working position, being supported by a weighing device capable of measuring the force exerted upon the vanes by the steam jet. From this force, with due regard to the angles of entrance and of exit, may be calculated the apparently realized mean velocity of the jet; but the method is hardly good for more than rough approximations, and the results must be used with care.

In Thomas' *Steam Turbines* diagrams are given showing a large number of observations of this kind: but they are much diminished in value by the fact that the nozzles used were not adapted to the conditions of working. With atmospheric pressure as p_2 and with p_1 varying from 30 to 110 lb. abs., the nozzles were all cylindrical but one, and that had a degree of divergence so large that it would be necessary to raise p_1 to more than 400 lb. in order to avoid overexpansion in the nozzle. The ratio of actual velocity (calculated from observed impulse) to ideal velocity ranges from 0.65 to 0.80, which corresponds with an energy efficiency of 0.40 to 0.64. If the nozzles had been proportioned to the pressure ratio, these results might have been considerably improved upon.

Stodola (Figs. 47 to 50 of the Second Edition) gives some curves showing the reactions observed with a similar arrangement; but in failing to supply quantitative data as to size of nozzle, rate of flow, etc., makes evident that he considers them merely of illustrative value.

(*p*) IMPULSE OF JET LEAVING VANES. — In the experiments of Briling the arrangement was as outlined at I in Fig. 338. A group of vanes was held fixed in front of the nozzle, and the flat measuring plate was set squarely across the current leaving the vane channels. This gave an impulse which was compared with that got by setting the plate in front of the nozzle itself, the ratio serving as a velocity coefficient,

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which in terms of the velocity diagrams in § 46 has the value

$$e_v = V_2 \div V_1. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (239)$$

The angles and spacing shown in Fig. 338, and found by test to give the best results, agree well with general practice. With these proportions, the coefficient E_v ranged from 0.50 to 0.75, rising by 0.10 to 0.12 with change from the smallest to the largest vanes tried. At II the dotted outline added to the first vane shows the profile that would be

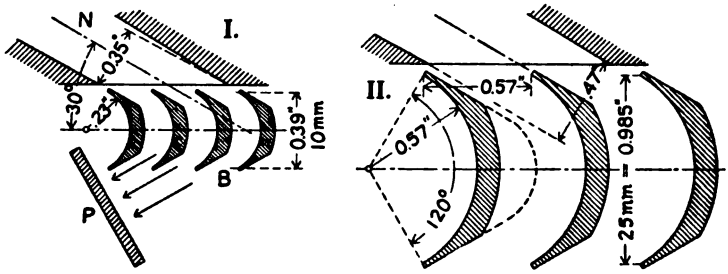

FIG. 338. — Smallest and Largest Blades used in Briling's Experiments — see reference on page 472: intermediate sizes had widths of 15 and 20 mm.

needed to give a channel of uniform width; while the actual vanes were made of brass plate, bent to curve and ground off at the edges. Varying the spacing or pitch, it was found that e_v grew less with change in either direction from the setting in Fig. 338, where pitch equals radius of curvature. Greater loss with closer spacing was attributed to more friction on the relatively larger amount of guiding surface; but it looks as if relatively greater width of the pocket between the vanes would be an even stronger influence toward wasteful action. With wider pitch there was, naturally, a poorer control and less definite deflection of the current.

Other vane profiles were tried, with entrance and exit angles of 20, 40, and 50 degrees, beside the 30° angles in Fig. 338: the swing or deflection of the current is less as the edges are more inclined, being 140° with 20° edge angles, only 80° with 50° edges. The smaller the angle of deflection, the less was the loss of velocity, ratio e_v going above 0.80 with 50° vanes. Over the range of experiment, with steam velocities of 250 to 1400 ft. per sec., the coefficient e_v increases slightly with rising steam velocity.

From data in the first part of Table 20, it may be estimated that the best running efficiency of the turbine wheel represented by the conditions of these experiments might be about 0.45 — this being, for the combination of nozzle, vanes, and wheel, the ratio of useful output to

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energy ideally available. The square root of 0.45 is 0.67, and is comparable with E_v; whence we see that Briling's results check up fairly well as to order of magnitude; still, their value is qualitative rather than quantitative.

(q) Energy Losses in the Turbine. — The total deficiency from ideal output is made up of losses in jet formation and in jet application, of residual kinetic energy in the jet when it leaves the vanes, and of friction work. The first item, jet formation, is most open to investigation, as may be inferred from the amount of space that has just been given to nozzle action. Loss in jet application, through steam friction, steam shock, stray motion, and leakage, is decidedly the largest item; but the possibility of following it out in detail in the running machine is very slight. Loss by residual energy is generally made quite small, and can be fairly well estimated or inferred. Friction includes both resistance to spinning of the rotor in the steam atmosphere which surrounds it and machine resistance in the bearings: its work absorption may be found by motor-driving the turbine or, in some cases, by shutting off steam and observing how rapidly the rotor loses speed because of friction, all external load having been taken off.

From the preceding statement it appears that in an analysis of turbine action the larger part of the energy loss must be determined by subtracting several smaller parts from the total loss. The latter is easily found, being one of the principal results from any performance test. Whether from the view-point of detailed analysis then, or with the simpler and more useful purpose of showing what the turbine can do, the presentation of test results is now in order.

§ 48. Turbine Performance

(a) Scheme of Table 20. — This collection of turbine data is generally similar to that for the engine in Table 13, pages 268 to 271, although some changes in the form of the quantities have been made. To save the trouble of referring back, all the symbols will now be briefly defined, with fuller explanation of points peculiar to the turbine.

Pages A, D. Description, Authority, Reference, etc.

First two groups (small turbines) and last group (low-pressure turbines) are determined by a controlling condition of operation. Between these the arrangement is by type: single-impulse wheels (Rateau, Zoelly, etc.); several-impulse wheels (Curtis, Schulz); combination of these two types (A.E.G. = Allgemeine Elektrizitäts Gesellschaft); reaction turbines (Parsons and variations); combined impulse and reaction types.

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TABLE 20, page A. TESTS OF VARIOUS STEAM TURBINES.

No.	Size, Make, Authority, Reference, etc.
	SMALL NONCONDENSING TURBINES.
1	30 h.p. De Laval, E. Lewicki, Dresden, 1901; Z.V.D.I., 1903, Vol. 47 I, 494.
2	150 h.p. Kerr, Page 286 }
3	25 h.p. Terry, Page 293 } Paper, "Small Steam Turbines," G. A.
4	50 h.p. Curtis, Page 285 } Orrok, Trans. A.S.M.E., 1909, Vol. 31,
5	200 h.p. Curtis, Page 285 } 263–310.
	SMALL CONDENSING TURBINES.
6	30 h.p. De Laval, E. S. Lea, Trans. A.S.M.E., 1904, Vol. 25, 1069.
7	50 h.p. De Laval, T. B. Morley, Glasgow, 1905; *Engineering*, 1905 II, Vol. 80, 880.
8	100 h.p. De Laval, Lodz, 1901; Z.V.D.I., 1901, Vol. 45 II, 1678.
9	300 h.p. De Laval, Dean & Main, Trenton, 1902; *Engineering Record*, 1902 II, 100.
10	20 h.p. Elektra, A. Stodola, 1906; }
11	200 kw. Elektra, Builders; } H. Meuth, Z.V.D.I., 1908, Vol.
12	200 kw. Elektra, Prof. Hubert; } 52 I, 182, 216; also STODOLA IV, 374.
13	24 kw. Parsons, Newcastle; }
14	50 kw. Parsons, Blackpool; } Paper, "Trials of Steam Turbines
15	100 kw. Parsons, Winwich; } Driving Dynamos," C. A. Parsons
16	200 kw. Parsons, Blackpool; } and G. G. Stoney, Int. Eng. Congress, 1901; *Engineering*, 1901 II, Vol. 72, 339.
	LARGE ONE-STAGE TURBINE.
17	1400 kw. Riedler-Stumpf, Moabit Station, Berlin, 1902, 3; F. Rötscher, Z.V.D.I., 1907, Vol. 51 I, 605, etc.; see STODOLA also.
	SINGLE-IMPULSE TURBINES.
18	500 h.p. Rateau, A. Stodola, Paris, 1903; STODOLA II E, 267; IV, 428.
19	1000 kw. Rateau, Oerlikon Works, 1903; STODOLA II E, 261.
20	1000 kw. Rateau, Oerlikon Works, recent; STODOLA IV, 435.
21	1500 kw. De Laval, Varberg, Sweden; *Power*, May 3, 1910.
22	300 kw. Zoelly, 3000 r.p.m. }
23	700 kw. Zoelly, 3000 r.p.m. } Z.V.D.I., 1910, Vol. 54 I, 330; STODOLA IV,
24	1000 kw. Zoelly, 3000 r.p.m. } 421, Table 4.
25	2300 kw. Zoelly, 1500 r.p.m. }
26	5000 kw. Zoelly, 1000 r.p.m., Reinisch-Westfälischen Elektricitätswerk, 1908; *Engineering*, 1908 II, Vol. 86, 1; STODOLA IV, 420.
27	3500 kw. Zoelly, 1500 r.p.m.; Alsace Mach. Co., Mulhouse; Z.V.D.I., 1910, Vol. 54 II, 1500.
	SEVERAL-IMPULSE TURBINES.
28	500 kw. Curtis, 1800 r.p.m., G. H. Barrus, Newport, 1904; Bulletin.

A.S.M.E., American Society of Mechanical Engineers. Up to Vol. 31, 1909, references are to the completed *Transactions;* later references are to the monthly *Journal.*
Electrical World, New York.
Engineering, London.
Engineering Record, New York.
Power (*and the Engineer*), New York.
STODOLA. Stodola's *Steam Turbines;* II E, Second Edition, in English, New York, 1905; IV, Fourth Edition, German, Berlin, 1910.
Z.V.D.I., *Zeitschrift des Vereines deutscher Ingenieure.*
Zeitschrift für gesämte Turbinenwesen.

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TABLE 20, page A — *Continued.*

No.	Various Notes.
1	One wheel, 8.9 in. diameter, gear ratio about 10 to 1, vane speed 775 ft. per sec. at 2000 r.p.m. of power shaft. Superheat run to very high value, above range of practice.
2	Eight impulse wheels, 24 in. diameter, vanes speed 293 ft. per sec. at 2800 r.p.m.
3	One 3- or 4-impulse wheel.
4	One 3-impulse wheel, 25 in. diameter, vane speed 400 ft. per sec. at 3600 r.p.m.
5	One 3-impulse wheel, 36 in. diameter, vane speed 314 ft. per sec. at 2000 r.p.m. The paper from which tests 2 to 5 are taken gives a good general description and illustration of a number of small turbines.
6	Machine similar in proportions to No. 1. See statement under Fig. 18, page 24, as to range of speed in De Laval turbines.
10	Machine has one four-impulse wheel — see Fig. 352.
11	Two three-impulse stages on one wheel, as in Fig. 352.
17	Largest single-stage turbine ever built, and the only one of its kind. See Fig. 354 for form of buckets, which are of the tangential type: this particular machine has but one velocity stage. Wheel is 2 m. or 78.8 in. in diameter, and bucket velocity is 1030 ft. per sec. at 3000 r.p.m.; machine has been run at 3800 r.p.m., as represented in Fig. 347.
18	Same machine as is shown in Fig. 20; 24 one-impulse wheels, 20 to 33 in. diameter, vane speed 220 to 345 ft. per sec. at 2400 r.p.m.
21	This machine is of the multistage type, generally similar to the Rateau and Zoelly turbines.
22	The construction of this turbine is fully described in both the references given. See partial drawing in Fig. 365.

Essential characteristics are very briefly stated; descriptive data vary so much in fulness that tabular columns cannot be made out for such things as wheel diameters and vane speed. For descriptions of form, refer to the illustrations in Chapters I and X, also to books on the turbine.

PAGES B, E. OPERATING CONDITIONS AND STEAM CONSUMPTION.

No. A serial number is given to each turbine; decimal figures denote different tests of the same turbine.

Old No. This is the number of the particular test in the original report. Sometimes several original tests are here averaged together.

(*Go to page* 483)

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TABLE 20, page B. TESTS OF VARIOUS STEAM TURBINES.

No.	Old No.	N	H, K	p_1	t_1	s	p_0	S_k	E_s	S_h	q_0
1.1		2014	44h	99.1	327	0	14.5	....	....	39.53	179.3
2		2354	25h	99.1	654	327	14.5	....	....	33.85	179.1
3		2096	51h	99.3	931	604	14.7	....	....	25.68	179.9
2.1		1000	100h	190	378	0	14.7	....	....	62.3	180
2		1600	140h	190	378	0	14.7	....	....	45.8	180
3		2800	180h	190	378	0	14.7	....	....	37.3	180
3.1	6	1800	23h	105	377	46	14.7	....	....	47.1	180
2	2	2100	24h	105	381	50	14.7	....	....	44.0	180
3	1	2500	26h	105	394	63	14.7	....	....	41.9	180
4		3600	50h	165	366	0	14.7	....	....	32.0	180
5		2000	200h	165	366	0	14.7	....	....	28.0	180
6		2000	30h	140	353	0	2.15	....	....	22.0	96.8
7		1635	48h	187	376	0	1.59	....	....	21.9	95.8
8		1076	242h	186	407	31	2.28	....	....	17.2	99.0
9.1		751	119h	216	388	.022	0.84	....	....	16.77	64.1
2		747	333h	221	390	.022	1.67	....	....	15.51	87.6
3		750	352h	222	475	84	1.47	....	....	13.94	83.1
10		3940	12	182	505	131	1.28	44.9	*.85*	28.40	78.2
11		2990	205	147	357	0	1.42	26.3	*.89*	17.42	81.9
12		3000	202	201	554	172	1.18	22.44	*.89*	14.91	75.4
13		4990	25	95	324	0	0.59	28.8	*.86*	18.47	52.8
14		5044	53	141	354	0	0.98	28.0	*.87*	18.18	69.2
15		3640	119	115	422	84	0.59	24.3	*.88*	15.95	52.8
16		3028	229	140	353	0	0.98	22.0	*.89*	14.61	69.2
17.1	12	965	210h	66	389	89	0.82	....	....	42.7	63.3
2	14	1770	1114h	183	503	128	1.35	....	....	21.35	80.1
3	5	2941	555	116	523	185	1.46	22.13	*.90*	14.86	82.8
4	7	2993	1334	205	554	170	2.19	19.80	*.94*	13.89	97.6
18.1	4	2184	108	176	376	5	1.29	30.42	.84	19.06	78.5
2	8	2101	366	169	388	20	1.64	22.60	.924	15.57	87.0
3	12	2360	463	224	415	24	2.15	22.09	.933	15.38	96.8
19	5	1500	1024	179	373	.005	2.43	21.98	*.95*	15.58	101.4
20	1	2613	1050	161	581	217	1.18	17.23	*.95*	12.21	75.4
21		1500	1570	182	542	168	0.75	16.47	.94	11.54	60.4
22	3, 4	3020	327	126	611	266	0.49	16.38	.912	11.14	47.0
23	4	3000	720	203	582	199	0.62	15.09	.920	10.36	54.3
24	3, 4	3000	1051	164	570	205	0.70	14.67	.916	10.02	58.2
25.1	1	1504	1120	198	600	219	0.59	15.44	.905	10.42	52.8
2	2	1509	1621	205	604	220	0.72	14.60	.926	10.10	59.1
3	3	1501	2508	198	607	226	0.98	14.11	.950	10.02	69.2
26	2, 3	1023	5014	170	531	163	1.67	16.16	.953	11.47	87.4
27.1		1500	1750	182	544	170	0.53	14.41	*.90*	9.68	49.5
2		1500	3500	183	514	140	0.79	14.09	*.95*	9.98	62.1
28.1		1815	529	165	366	0	1.00	19.78	*.93*	13.71	69.8
2		1815	515	165	656	290	1.00	15.91	*.93*	11.05	69.8

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TABLE 20, page C. TESTS OF VARIOUS STEAM TURBINES.

h_1	h_2	Q	W_R	W_h	W_k	E_h	E_{Rh}	E_{Rk}	Q_{mh}	No.
1186	1046	1007	140.0	64.4	....	.064	.457	....	665	1.1
1353	1166	1173	186.9	75.2	....	.064	.402	....	663	2
1488	1255	1308	232.6	95.5	....	.073	.411	....	582	3
1198	1012	1018	185.8	40.8	....	.040	.220	....	1055	2.1
1198	1012	1018	185.8	55.6	....	.065	.299	....	656	2
1198	1012	1018	185.8	68.3	....	.067	.370	....	628	3
1214	1065	1034	148.2	54.1	....	.052	.365	....	812	3.1
1216	1065	1036	150.7	57.8	....	.056	.384	....	761	2
1223	1073	1043	149.4	60.7	....	.058	.407	....	730	3
1195	1019	1015	176.2	79.5	....	.078	.451	....	543	4
1195	1019	1015	176.2	90.9	....	.089	.515	....	465	5
1193	916	1096	277.0	115.7	....	.106	.418	....	402	6
1198	889	1102	308.5	116.2	....	.105	.377	....	402	7
1217	921	1118	296.2	148.0	....	.133	.499	....	318	8
1181	867	1120	314.0	151.8	....	.135	.483	....	313	9.1
1182	872	1094	309.4	164.1	....	.150	.531	....	283	2
1247	911	1163	335.8	182.4	....	.157	.543	....	270	3
1273	928	1194	344.7	89.6	76.0	.075	.260	.220	565	10
1193	895	1112	298.4	145.9	129.7	.131	.488	.434	234	11
1297	930	1222	366.9	170.7	151.8	.140	.465	.413	303	12
1186	872	1133	313.7	137.9	118.5	.122	.440	.378	348	13
1193	880	1124	313.2	140.0	121.9	.125	.447	.389	340	14
1236	894	1183	341.9	159.5	140.3	.135	.467	.411	314	15
1193	880	1124	312.8	174.0	155.1	.155	.556	.495	274	16
1225	936	1162	289.5	59.6	....	.051	.206	....	828	17.1
1271	930	1191	341.4	119.2	....	.100	.350	....	424	2
1288	971	1205	316.6	171.3	154.2	.142	.541	.487	299	3
1296	963	1199	302.8	183.5	172.2	.153	.605	.568	277	4
1200	885	1121	315.1	133.6	112.0	.119	.423	.355	356	18.1
1209	905	1122	303.8	163.5	150.9	.146	.538	.497	291	2
1217	907	1120	310.2	165.5	154.4	.148	.533	.498	286	3
1197	913	1095	284.2	163.6	155.2	.149	.575	.546	284	19
1312	954	1236	357.5	208.3	198.0	.169	.583	.554	252	20
1292	913	1231	378.9	220.3	207.0	.179	.582	.547	237	21
1333	932	1286	401.1	228.6	208.4	.178	.570	.520	238	22
1310	906	1256	404.2	245.9	226.1	.196	.608	.560	217	23
1308	923	1249	384.4	254.2	232.6	.204	.661	.605	208	24
1320	910	1268	410.0	244.3	220.9	.193	.596	.539	220	25.1
1322	919	1263	403.0	252.1	233.7	.200	.626	.580	212	2
1324	938	1255	386.2	254.5	241.8	.203	.659	.625	209	3
1287	954	1200	332.9	222.0	211.2	.185	.667	.635	229	26
1293	896	1243	397.0	263.0	236.6	.211	.663	.596	201	27.1
1278	906	1216	371.2	255.0	242.1	.210	.688	.653	202	2
1195	872	1126	323.4	185.6	172.5	.165	.573	.533	257	28.1
1355	963	1285	392.5	230.2	214.2	.179	.587	.546	237	2

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TABLE 20, page D. TESTS OF VARIOUS STEAM TURBINES — *Continued.*

No.	Size, Make, Authority, Reference, etc.
	TURBINES OF CURTIS TYPE — *Continued.*
29	1000 kw. Thomson-Houston Curtis, 1000 r.p.m.; *Engineering*, 1907 II, Vol. 84, 375.
30	5000 kw. Curtis, 750 r.p.m., Boston, 1907;
31	8000 kw. Curtis, 750 r.p.m., Chicago, 1907; } See *Power*, Dec. 20, 1910, 2251; also next page.
32	8000 kw. Curtis, 750 r.p.m., New York, 1908;
33	9000 kw. Curtis, 750 r.p.m., Oakland, Cal., 1910; F. H. Varney, Jour. A.S.M.E., Dec. 1910, Vol. 32, 2089.
34	450 kw. Schulz, M. F. Gutermuth, 1909; Z.V.D.I., 1910, Vol. 54 I, 82.
	MIXED-TYPE IMPULSE TURBINES.
35	4000 kw. A.E.G. (German General Electric), Moabit Station, Berlin; STODOLA IV, 399.
36	4000 kw. A.E.G., Rummelsburg Station, Berlin, 1908; *Zeit. für ges. Turbinenwesen*, May 10, 1909; Z.V.D.I., 1909, Vol. 53 I, 761; STODOLA IV, 404.
	REACTION TURBINES, PARSONS AND VARIATIONS.
37	400 kw. Westinghouse-Parsons, 3500 r.p.m., Dean & Main, Pittsburgh, 1903; F. Hodgkinson, Trans. A.S.M.E., 1904, Vol. 25, 716.
38	5000 kw. Brown-Boveri-Parsons, 1000 r.p.m.; Essen, 1907; STODOLA IV, 449.
39	7500 kw. Westinghouse-Parsons, 750 r.p.m.; New York, 1907; *Electrical World*, Dec. 14, 1907.
40	4000 kw. Allis-Chalmers-Parsons; 1800 r.p.m.; R. C. Carpenter, Richmond, Va., 1908; STODOLA IV, 467; also *Sibley Journal of Engineering*, Jan., 1911.
41	3500 kw. Brown-Boveri-Parsons, 1350 r.p.m.; Frankfort; Z.V.D.I., 1908, Vol. 52 I, 516; STODOLA IV, 449.
42	3500 kw. Parsons, 1200 r.p.m.; Carville Station, Newcastle-on-Tyne; *Engineering*, 1907 I, Vol. 83, 654; STODOLA IV, 439.
43	6000 kw. Brown-Boveri-Parsons, 1200 r.p.m.; Mertz & McLellan, Newcastle-on-Tyne; *Engineering*, 1911 I, Vol. 91, 314.
	MIXED-IMPULSE AND REACTION TURBINES.
44	500 kw. Melms-Pfenninger, 2500 r.p.m.; M. Schroeter, Munich, 1906; Z.V.D.I., 1906, Vol. 50 II, 1811, etc., STODOLA IV, 481.
45	2400 kw. Sulzer, 1500 r.p.m.; Paris; STODOLA IV, 475.
46	10,000 kw. Westinghouse Double-flow, 1800 r.p.m., San Francisco, 1910; S. T. Naphtaly, Jour. A.S.M.E., Dec., 1910, Vol. 32, 2105.
	LOW-PRESSURE TURBINES.
47	300 h.p. Rateau, 1600 r.p.m.; Bruay, 1902; A. Rateau, Trans. A.S.M.E., 1904, Vol. 25, 817; STODOLA II E, 263.
48	500 kw. Rateau, 1500 r.p.m.; Hallside Works, 1906; *Engineering*, 1906 I, Vol. 81, 848.
49	600 kw. Brush-Parsons, 2000 r.p.m.; *Engineering*, 1910 II, Vol. 90, 8.
50	5000 kw. Curtis, 720 r.p.m.; New York, 1909–10; H. G. Stott, R. J. S. Pigott, Jour. A.S.M.E., Mar., 1910, Vol. 32, 315.

NOTE. — The heat quantities h_1, h_2, and Q in this table are the nearest integral approximations to calculated values with one decimal place.

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TABLE 20, page D — *Continued.*

No.	Various Notes.
30 31 32	The *Power* reference contains summary of turbine tests, by Mr. G. A. Orrok, presented in discussion of A.S.M.E. papers describing tests of turbines here listed as Nos. 33 and 46. Only a few leading results from Nos. 30 to 32 have been published, in scattered fashion. The test of No. 31 was very complete; perhaps the best summary of what has been made public will be found in *Engineering*, 1907 II, Vol. 84, 111. These are, presumably, all 5-stage machines; speed is either 750 r.p.m. with 25-cycle current or 720 r.p.m. with 60-cycle current.
34	Horizontal machine, four 2-impulse wheels about 45 in. diameter; vane speed 435 ft. per sec. at 2200 r.p.m. See Fig. 348 for diagram and analysis of second test, and § 51 (*o*) for remark on scheme of regulation.
35 36	Large turbines of this make have one or two Curtis wheels (2-impulse) followed by a series of single-impulse wheels.
43	This turbine is made in two sections, like that shown in Fig. 20; low-pressure staging is especially complete, for full utilization of high vacuum.
44	A series of single-impulse vane rows, mounted on the rotor drum instead of on wheels (see Fig. 355), is followed by reaction stage groups of the typical Parsons form.
45	The general scheme of this turbine is well illustrated by Fig. 26.
46	See Fig. 357 for general drawing of turbine.
47	The turbine was installed for operation in conjunction with a Rateau accumulator — see § 50 (*f*) — but the test was evidently run with steam throttled from high pressure, as shown by the recorded superheat.
48	Exhaust from several high-pressure engines was collected in an accumulator, where pressure ranged from 15 to 17 lb. absolute. As given in the first three tests, p_1 is the pressure below the governor valve, or at entrance to the first set of nozzles. Test 4 is the same as No. 3, but is worked out for the initial pressure above the throttle valve, with steam of the same initial total heat. Machine has eleven wheels, about 40 in. diameter; vane speed 260 ft. per sec. at 1500 r.p.m.
49	Turbine fully described and illustrated in reference; double-flow arrangement, steam from ends to middle; mean vane-row diameter about 27 in., velocity about 240 ft. per sec. at 2000 r.p.m.
50	Three two-impulse wheels, regular vertical turbine, condenser in base. Turbine in series with large engine, listed as No. 27 in Table 13. Only a separator (not very effective, apparently) between engine and turbine, so that pressures of engine exhaust and of turbine admission vary together. Very full data in paper, but see note under Table 25. Tests 1 to 4 represent series in which pressure between engine and turbine varied with load; while in tests 5 to 7 this pressure was held practically constant.

(*From page* 479)

N Speed in revolutions per minute: in many cases only the nominal speed is given.

H Brake or effective horse-power: with the smaller machines, the load is expressed (and was measured) in this unit; such values, in the column under *H* and *K*, are marked by the letter h.

Indicated horse-power, to be defined by analogy to the engine as the rate of work performance by the steam upon the vanes, is an imaginable quantity for the turbine; it may be calculated

(*Go to page* 486)

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TABLE 20, page E. TESTS OF VARIOUS STEAM TURBINES.

No.	Old No.	N	H, K	p_1	t_1	s, m	p_0	S_k	E_e	S_h	q_0
29.1		1000	498	168	384	16	0.47	19.4	*.90*	13.03	45.8
2		"	1005	168	389	21	0.41	17.0	*.94*	11.92	41.7
3		"	1005	168	466	98	0.41	15.9	.94	11.15	41.7
4		"	1275	163	508	143	0.47	15.2	*.95*	10.78	45.8
30		750	5194	180	515	142	0.59	13.52	*.97*	9.79	52.8
31		750	10816	190	525	147	0.25	12.90	*.97*	9.34	37.3
32		750	8880	193	487	109	0.93	15.05	*.97*	10.90	67.4
33		750	8775	195	453	74	0.97	15.95	.97	11.53	68.8
34.1	14	2200	83	194	539	160	1.41	29.67	.76	16.81	81.6
2	29	2183	447	196	543	163	1.39	20.40	.905	13.77	81.1
3	1*	2104	402	188	377	0	1.25	22.93	.905	15.47	77.4
35		1500	3206	190	646	268	0.68	12.71	*.95*	9.01	57.0
36.1	1, 2	1496	2218	198	643	262	0.28	11.85	*.93*	8.22	30.5
2	3, 4	1500	3243	196	629	249	0.28	12.07	*.95*	8.55	30.5
3	5, 6	1497	4235	194	654	275	0.38	12.08	*.96*	8.65	39.4
37.1		3549	594h	170	368	0	1.46	....	....	14.35	82.8
2		3545	594h	169	368	0	0.99	....	....	13.91	79.5
3		3544	593h	165	470	104	0.98	....	....	12.50	79.2
4		3543	592h	161	544	180	0.93	....	....	11.46	67.4
38		1000	5128	176	565	194	0.75	14.32	.94	10.05	60.4
39		750	9865	191	474	96	1.32	15.11	.97	10.94	79.3
40.1		1800	1393	180	450	77	0.82	17.26	*.92*	11.85	63.3
2		"	4328	186	484	108	0.84	14.00	*.96*	10.03	64.1
41		1350	3522	162	498	133	0.55	13.70	*.95*	9.70	50.6
42.1	3	1200	2193	217	492	103	0.47	14.52	*.925*	10.03	45.8
2	7, 8	"	5112	212	493	106	0.43	13.30	*.95*	9.43	43.1
3	5	"	6222	211	503	117	0.51	13.46	*.955*	9.60	48.3
43.1		1200	4256	205	546	162	0.40	12.42	*.945*	8.76	51.0
2		"	5600	204	550	166	0.43	12.12	*.955*	8.65	53.1
3		"	6257	204	560	176	0.44	11.95	*.96*	8.56	53.8
44.1	4	2489	147	182	583	209	0.37	22.50	.79	13.28	48.6
2	1	2459	499	189	607	230	0.50	17.18	.905	11.60	47.7
45.1		1500	1200	171	569	200	0.62	18.33	.91	12.46	54.3
2		"	2400	171	569	200	0.62	15.28	.93	10.60	54.3
46.1	1	1800	5333	188	431	54	0.82	15.66	*.95*	11.10	63.3
2	5	"	9173	182	433	59	1.03	14.57	*.97*	10.54	70.8
47	5	1598	233	14.7	297	85	2.79	39.97	*.94*	28.01	106.7
48.1	1	1500	69	2.90	140	*.000*	0.58	66.4	*.84*	41.65	52.3
2	5	"	241	6.11	171	*.012*	0.74	39.3	*.92*	27.00	60.0
3	18	"	450	11.40	199	*.024*	0.98	36.6	*.94*	25.68	69.2
4	18	"	450	14.70	212	*.029*	0.98	36.6	*.94*	25.68	69.2
49.1		2000	306	14.7	212	10	0.52	35.2	*.92*	24.19	48.9
2	Mn.	"	600	14.7	212	15	0.87	31.8	*.95*	22.54	55.2
50.1	58	750	2213	7.08	177	.043	0.48	36.85	*.92*	25.30	46.4
2	60	"	4860	12.10	202	.027	0.43	29.20	*.96*	20.93	43.1
3	54	"	6283	15.18	214	.031	0.46	28.11	*.965*	20.22	45.1
4	38	"	7784	20.60	222	.098	0.74	30.51	*.97*	22.08	60.0
5	44	"	4426	16.10	217	.079	0.46	27.72	*.958*	19.80	45.1
6	42	"	4938	16.24	217	.070	0.65	29.80	*.96*	21.35	55.8
7	39	"	5895	16.50	218	.106	0.98	31.75	*.965*	22.86	61.7

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TABLE 20, page F. TESTS OF VARIOUS STEAM TURBINES.

h_1	h_2	Q	W_R	W_h	W_k	E_h	E_{Rh}	E_{Rk}	Q_{mh}	No.
1206	844	1161	362.8	195.2	175.9	.168	.538	.485	252	29.1
1210	840	1168	369.5	213.5	200.5	.183	.578	.543	232	2
1253	866	1212	387.4	228.2	214.5	.188	.589	.554	226	3
1276	887	1230	389.0	236.2	224.5	.192	.607	.577	221	4
1278	894	1225	384.6	260.2	252.3	.213	.677	.656	199	30
1282	853	1245	429.2	272.5	264.4	.219	.635	.616	194	31
1262	903	1195	359.1	233.9	226.7	.196	.652	.631	217	32
1234	892	1175	351.3	220.8	213.9	.188	.629	.610	226	33
1289	939	1208	350.2	151.3	115.0	.125	.432	.328	339	34.1
1291	939	1210	352.8	184.8	167.2	.153	.524	.474	277	2
1198	877	1120	320.5	164.6	149.0	.147	.513	.464	288	3
1343	931	1286	412.0	282.4	268.3	.220	.685	.652	193	35
1342	885	1311	457.0	309.4	288.0	.236	.677	.631	180	36.1
1335	882	1305	452.7	297.6	282.7	.228	.658	.625	186	2
1347	903	1308	444.3	294.0	282.5	.225	.662	.636	189	3
1196	889	1113	306.5	172.4	....	.155	.563	....	274	37.1
1196	871	1116	325.1	183.0	....	.164	.563	....	259	2
1256	911	1177	344.6	203.7	....	.173	.591	....	245	3
1295	932	1230	363.0	222.3	....	.181	.613	....	234	4
1304	921	1240	383.4	253.3	238.2	.204	.660	.621	208	38
1255	916	1176	339.4	232.7	225.9	.198	.685	.665	214	39
1243	888	1180	354.6	215.0	197.8	.182	.606	.558	233	40.1
1262	899	1198	363.4	254.0	243.9	.212	.699	.671	200	2
1271	892	1220	378.8	262.1	249.0	.215	.692	.657	197	41
1262	863	1217	399.4	253.9	234.8	.209	.636	.589	213	42.1
1263	861	1220	402.8	270.0	256.6	.221	.670	.637	192	2
1269	872	1221	396.8	265.0	253.2	.217	.668	.637	195	3
1292	875	1241	416.5	290.2	274.5	.234	.698	.659	181	43.1
1294	879	1243	414.7	294.2	281.3	.237	.710	.679	179	2
1299	883	1246	416.0	297.5	285.5	.239	.715	.686	177	3
1313	889	1264	423.8	191.7	151.6	.152	.453	.358	280	44.1
1324	907	1277	417.7	219.3	198.6	.172	.525	.475	247	2
1307	914	1252	392.9	204.3	186.0	.163	.520	.473	260	45.1
1307	914	1252	392.9	240.0	223.2	.192	.612	.568	221	2
1232	879	1168	352.7	229.2	218.0	.196	.650	.618	216	46.1
1233	892	1163	340.9	241.3	234.0	.208	.708	.687	204	2
1190	1074	1083	116.2	90.8	85.4	.084	.782	.735	505	47
1121	1024	1069	97.2	61.1	51.3	.057	.628	.528	740	48.1
1121	994	1061	126.9	94.3	86.8	.079	.745	.685	537	2
1121	971	1052	149.9	99.1	93.2	.094	.661	.622	452	3
1122	937	1052	184.9	99.1	93.2	.094	.536	.504	452	4
1135	960	1086	174.7	105.2	96.9	.097	.602	.555	438	49.1
1135	959	1080	175.9	113.0	107.3	.105	.642	.610	405	2
1093	937	1046	156.0	100.6	92.5	.096	.645	.593	442	50.1
1120	924	1077	195.7	121.7	116.8	.113	.622	.597	375	2
1120	915	1075	205.3	125.8	121.3	.117	.613	.592	362	3
1062	875	1002	187.0	115.2	111.8	.115	.617	.598	369	4
1055	876	1010	179.0	128.5	123.1	.127	.718	.688	334	5
1084	899	1028	184.8	119.2	114.5	.116	.645	.620	366	6
1050	888	988	161.3	111.3	107.5	.113	.690	.667	377	7

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(*From page 483*)

by adding the friction work (both wheels-in-steam and bearing) as defined in § 47 (*q*) to the shaft output. This is not a practical determination, and is attempted only in special and highly detailed experiments.

K Load in kilowatts; all the larger machines were electrically loaded, and all values in the load column are in kilowatts unless marked h as just noted.

p_1 Initial steam pressure, above the throttle or control valve of the turbine, in pounds per square inch absolute. In one low-pressure turbine, No. 48, tests 1 to 3, which received steam at or a little above atmospheric pressure from a Rateau accumulator — see § 50 (*f*) — the initial pressure given is that below the governor valve.

t_1 Initial steam temperature, measured at same place as p_1, degrees fahrenheit.

m Fraction of moisture in entering steam, if saturated.

s Degrees of superheat of entering steam.

p_0 Exhaust pressure, preferably and presumably measured in exhaust pipe of turbine; pounds per square inch absolute.

S_k Pounds of steam consumed per kilowatt-hour.

E_e Electrical efficiency of generator; when printed in italics, this has been assumed by author, at what seems a probable value.

S_h Pounds of steam consumed per shaft or effective horse-power-hour: as shown in § 28 (*e*), page 279,

$$S_h = S_k \times (0.746 \times E_e). \quad . \quad . \quad . \quad . \quad . \quad (240)$$

q_0 Heat in one pound of feed water, which is taken to be at the temperature corresponding (as for saturated steam) to the exhaust pressure p_0: see § 26 (*e*).

PAGES C, F. THERMODYNAMIC PERFORMANCE.

h_1 Total heat of one pound of steam at pressure p_1 and of the quality fixed by m or s.

h_2 Total heat at the end of adiabatic expansion from p_1 to p_0.

Q Input of heat per pound of steam, equal to $(h_1 - q_0)$.

W_R Output of the ideal Rankine cycle, in B.t.u. per pound of steam, equal to $(h_1 - h_2)$.

W_h Actual work output per pound of steam, expressed in B.t.u., based on effective horse-power, and equal to $(2545 \div S_h)$.

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W_k Work output per pound of steam, of turbine and generator combined, equal to $(3412 \div S_k)$. Evidently,

$$W_k \div W_h = E_e. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (241)$$

E_h Absolute thermodynamic efficiency, based on effective horse-power, therefore equal to $(W_h \div Q)$.

E_{Rh} Relative efficiency of turbine alone, equal to $(W_h \div W_R)$.

E_{Rk} Relative efficiency of turbine and generator, equal to $(W_k \div W_R)$.

Q_{mh} Heat supplied per effective horse-power per minute, equal to $(42.4 \div E_h)$.

(*b*) Discussion of Table 20. — Between turbines of different designs it is not well to draw close comparisons, unless much fuller data are available (as to both design and operation) than are generally made public — far more than are given in Table 20. It is thought, however, that the averaged results in Table 21 are valuable and instructive, as showing what has been attained by the best examples of the several types. The figures speak for themselves.

Table 21. Thermodynamic Efficiencies of Various Types of Turbines, with Generators, by Group Averages from Table 20.

Group.	Load. K	St. Pr. p_1	Supht. s	Ex. Pr. p_0	Efficiencies. E_h	E_{Rh}	E_{Rk}	E_e
A	2770	178	184	1.04	0.200	0.666	0.630	0.955
B	8420	176	118	0.66	0.204	0.647	0.628	0.97
C	3560	186	264	0.45	0.218	0.668	0.638	0.955
D	5700	193	136	0.73	0.215	0.687	0.656	0.955
E	9170	182	59	1.03	0.208	0.708	0.687	0.97

A. Zoelly turbines, single-impulse wheels, tests 24, 25.3, 26, 27.2.
B. Curtis turbines, two-impulse wheels, tests 30, 31, 32, 33.
C. A.E.G. (Curtis and Zoelly types combined), tests 35, 36.2, 36.3.
D. Parsons type, many-stage reaction, tests 38, 39, 40.2, 41, 42.2, 43.2.
E. Curtis and Parsons types combined, test 46.2.
The tests selected for averaging are those near the rated power of the machine.

For the same or similar turbines, it is of decided interest and importance to see how efficiency varies with such governing conditions as size, load, steam pressure, steam quality, exhaust pressure, and speed; and these matters will now be taken up seriatim. In making comparisons and forming judgment, the relative efficiency E_R is by far the most useful criterion.

(*c*) Influence of Size. — This is well represented by Fig. 339, which shows how the ratio of actual to ideal output varies with rated power. The curve, sketched in by eye, is intended to serve as an approximate mean of the open-circled points belonging to the turbine

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alone, or derived from brake or shaft output. Naturally, with all kinds of machines represented, there is wide variation of points from curve, and the latter must not be given too high a standing as a law of relation.

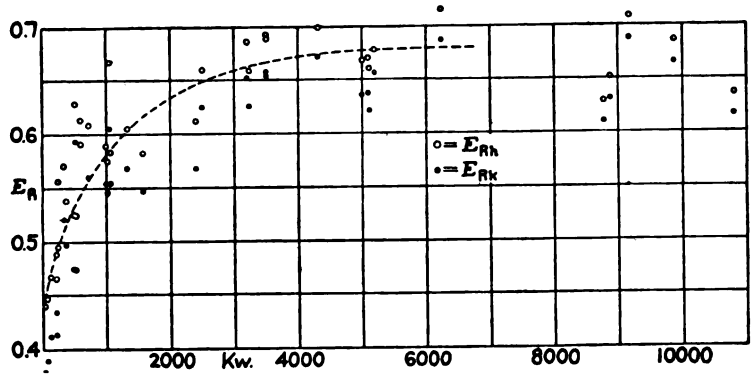

FIG. 339. — Influence of Size upon Relative Efficiency of Steam Turbines. Base electrical output K in kilowatts; ordinate, relative efficiency E_R, from turbine output as E_{Rh}, from generator output as E_{Rk}. Points from turbines 11 to 46 in Table 20, using tests at or near rated power.

One thing to be noted is that above 3000 to 4000 kw. capacity size apparently ceases to have any influence upon economy.

(*d*) INFLUENCE OF LOAD FACTOR. — Figure 340 shows how the relative efficiency falls off with decrease of load, but continues to increase as the load goes above rating. In the latter characteristic the turbine

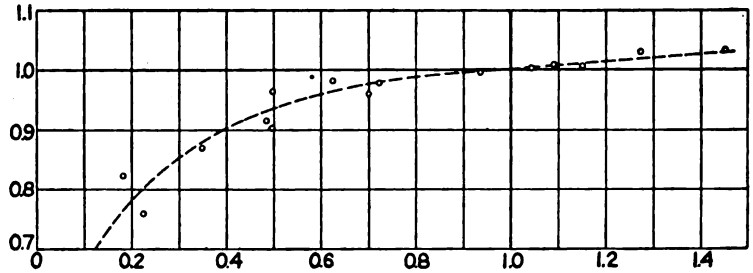

FIG. 340. — Variation of Efficiency with Load Factor. Base, ratio of actual electrical load to rated load; ordinate, ratio of relative efficiency E_{Rh} at the particular load to that at rated load. Data from turbines 25, 27, 29, 34, 40, 42, and 43 in Table 20.

differs from the piston engine, which at once begins to lose efficiency when overloaded — in the two types of machine rating bears about the same ratio to maximum capacity. The examples here represented are much fewer than in Fig. 339, and again the curve sketched in as a rough mean has little more than illustrative value. One turbine, No. 36,

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is intentionally omitted, because it follows the opposite law of an increase of E_R with decrease of load: this is due to the extremely low exhaust pressure, and will receive fuller comment in Art. (*h*).

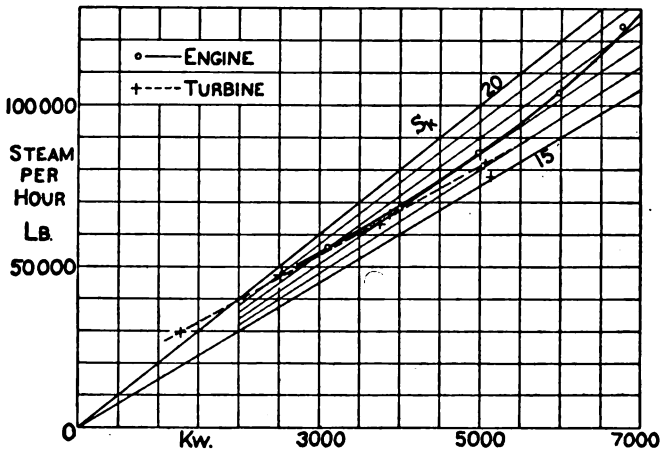

FIG. 341. — Comparison of an Engine and a Turbine, as to Variation of Steam Consumption with Load. Engine, 42 and 86 by 60 in. duplex compound, rated 5000 kw., No. 27 in Table 13, page 268. Turbine, 5000 kw. Zoelly, No. 26 in Table 20. Base is electrical output in kilowatts; ordinate shows total pounds of steam per hour to scale at left, also pounds per kilowatt-hour to scale formed by inclined lines — compare Fig. 87, etc., in Chapter V.

The contrast between characteristic steam-consumption curves for engine and for turbine is well illustrated in Fig. 341. The engine tests do not run low enough to give a good comparison at light loads; but whereas the two machines are in practical agreement from 3000 to 4000 kw., the steam rate S_k of the turbine continues to decrease as the load rises, while that of the engine passes a minimum and then increases quite rapidly.

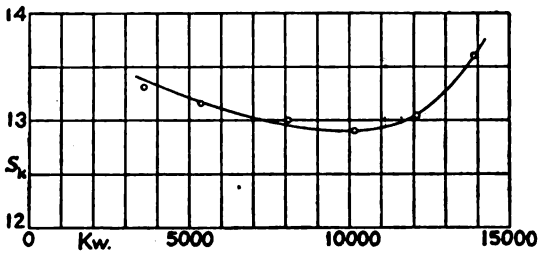

FIG. 342. — Steam Consumption of 8000 Kw. Curtis Turbine, No. 31 in Table 20.

Figure 342 is a simple plot of steam rate on output for a large turbine: it shows that S_k, after slowly falling to a minimum, rises rather abruptly with heavy overload. Just how such an overload is carried

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depends upon the method of governing, or upon the "valve gear" of the turbine — see § 52. The scheme of bypassing the first stages, throwing them more or less completely out of action, seems most likely to give a definite and decided rise in steam consumption at high loads.

(*e*) INFLUENCE OF STEAM CONDITIONS. — The three elements of initial pressure, initial quality, and exhaust pressure are closely linked together. Concerning these, experience with steam turbines has crystallized into certain general conclusions or opinions which have almost, if not quite, the standing of fundamental principles. After these have been stated, they will be discussed with the help of the data in Table 20.

1. The turbine works more effectively at the low-pressure than at the high-pressure end of the operation of expansion; in other words, the losses from ideal effect are greater with high steam density than with low density.

2. Superheat is very beneficial, diminishing steam friction and greatly reducing steam consumption.

3. The turbine can effectively receive and apply the energy made available by lowering the exhaust pressure, hence high vacuum is desirable and economical.

To give a general view of information along this line, the plot in Fig. 343 is made from Table 20, using rated-load results from the large

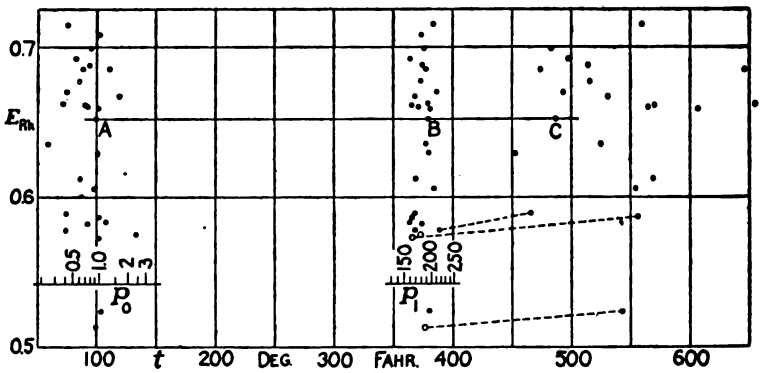

FIG. 343. — Efficiency in Conversion of Energy, as related to range of steam pressure and of superheat.

and medium-size turbines The relative efficiency by turbine output, E_{Rh}, is measured vertically, to the scale at the left. On each line like ABC thus located are laid off to the base scale, first, at A, the exhaust temperature t_0 as taken from Table I for p_0; next, at B, the saturation temperature corresponding to p_1; lastly, at C, the actual initial tem-

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perature. The two circled dots under B show tests in which there was no superheat.

As might be expected, with data from various kinds and sizes of turbines, there is no sign of any law of relation between the steam conditions and efficiency. It is therefore necessary to fall back upon the few cases in which conditions were varied with the same turbine. Two of these are given in the diagram, where the dotted inclined lines connect points from the same machine, and show moderate improvement as the result of superheating.

(*f*) The Question of Pressure Range. — Comparing the low-pressure or exhaust-steam turbines in Table 20 with those of full range, we note a decided advantage in relative efficiency for the low range. The stage efficiencies in the example in Table 24, Art. (*k*), although somewhat erratic in their manner of variation, also help to confirm statement 1 in the last article.

Low efficiency in the upper stages is readily accounted for by greater steam friction and by greater leakage. Experiments on rotor resistance have shown that surface friction is just about proportional to steam density. As to leakage, there is much more room for this in drum-rotor turbines than in those having a series of wheels in chambers separated by diaphragms. Some clearance between running and standing parts is always necessary, and the greater the diameter of the running joint the greater the opening for leakage. At the high-pressure end of a reaction turbine especially, clearance area will bear a high ratio to the small channel area between short vanes.

The poorer working of the turbine at high pressure is a strong argument for the combination of high-pressure engines with low-pressure turbines — see the results from a large combined unit set forth in Art. (*m*). The advantage is scarcely great enough to justify the higher cost of the double system in a new plant for electrical service; but it is all in favor of increasing the capacity and economy of existing engine plants by adding low-pressure turbines.

(*g*) Effect of Superheating. — A number of comparative tests without and with (or with varying degrees of) superheat are collected in Table 22; these are from small and medium-size turbines, since the tests of big machines have all been run with superheated steam, except in low-pressure operation. The actual gain in thermal efficiency E_h is not nearly so great as the apparent improvement in steam consumption S_h; but except where the type of construction limits capability of getting the full benefit of increased steam velocity (tests 1, 11, 12), there is a gain in relative efficiency E_{Rh}. No remarkable benefit from superheating is apparent; and it may well be questioned whether a

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TABLE 22. TURBINE TESTS SHOWING EFFECT OF SUPERHEATING.

No.	Load.	p_1	p_0	s	S_h	E_h	E_{Rh}	Type.
1.1	44 h	99	14.5	0	39.5	.064	.457	De Laval.
3	51 h	99	14.7	604	25.7	.073	.411	
9.2	333 h	221	1.67	0	15.5	.150	.531	De Laval.
3	352 h	222	1.47	84	13.9	.157	.543	
11	205 k	147	1.42	0	17.5	.131	.488	Elektra.
12	202 k	201	1.18	172	14.9	.140	.465	
28.1	529 k	165	1.00	0	13.7	.165	.573	Curtis.
2	515 k	165	1.00	290	11.1	.179	.587	
29.2	1005 k	168	0.41	21	11.9	.183	.578	Curtis.
3	1005 k	168	0.41	98	11.2	.188	.589	
4	1375 k	163	0.47	143	10.8	.192	.607	
34.3	402 k	188	1.25	0	15.5	.147	.513	Schulz.
2	447 k	196	1.39	163	13.8	.153	.524	
37.2	594 h	169	0.99	0	13.9	.164	.563	Parsons.
3	593 h	165	0.98	104	12.5	.173	.591	
4	592 h	161	0.93	180	11.5	.181	.613	

For meaning of symbols, see Art. (*a*).

maximum in overall economy, or in commercial efficiency, is not reached at a lower degree of superheat than is given in many of the tests in Table 20.

The absence of valves and pistons obviates most of the operating difficulties which arise from the use of superheated steam in the engine: there remain the danger of burning out the tubes of the superheater, more trouble with steam piping, and greater temperature distortion of the turbine. Against these lie the advantages of lower steam friction and (reported) less wear of vanes by dry, gaseous steam. As a summary of the examples in Table 22, and with other less purely technical data in mind, it may be stated that a thermal gain of 7 to 10 per cent (in absolute efficiency E_h) and a commercial gain of perhaps 4 to 6 per cent is about all that can be expected from the use of superheated steam: while with excessive superheat the cost efficiency of the whole plant (counting in depreciation and repairs) may fall below the maximum attainable.

In contrast with high-pressure practice, note how wet was the steam supplied to low-pressure turbine No. 50, Table 20: this was in spite of a separator intended to remove water from the engine exhaust.

(*h*) EFFECT OF VACUUM. — Of comparative tests in which exhaust pressure was the principal variable, but few can be selected from Table

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20; and of the first two pairs in Table 23, Nos. 26 x and 34 x are not given in the main table. Test 26 x appears on Fig. 341 in an abnormally low point, hence the apparent gain cannot all be credited to greater vacuum.

TABLE 23. TESTS SHOWING VARIATION OF VACUUM.

No.	Load.	p_1	s	p_0	S_h	E_h	E_{Rh}
26	5015	170	163	1.67	11.47	.185	.667
26 x	5141	166	176	1.20	10.75	.195	.674
34.2	447	196	163	1.39	13.77	.153	.524
34 x	396	191	189	0.80	13.20	.155	.501
37.1	594	170	0	1.46	14.35	.155	.563
2	594	169	0	0.99	13.91	.164	.563
36.1	2218	198	262	0.28	8.22	.236	.677
2	3243	196	249	0.28	8.55	.228	.658
3	4235	194	275	0.38	8.65	.225	.662

For meaning of symbols, see Art. (*a*).

The essential deduction from Table 23, from a wider view of test data, and from rational considerations, is that a particular turbine cannot follow, with undiminished relative efficiency, a lowering of the exhaust pressure below the value for which the machine was designed. Load and rate of steam flow remaining nearly the same, the last stage or stages will get more energy and higher steam velocities as the vacuum is greater; but with a fixed vane speed these higher velocities cannot be fully utilized.

If the efficiency ratio E_R were maintained constant by the machine, a plot of steam rate S_h on exhaust pressure p_0 would give a curve like AB, Fig. 344; but instead of thus falling off at an increasing rate as p_0 is less, the steam consumption per power unit follows very nearly a straight line like AC. If p_0 rises above the best value, line CA will be produced upward, showing a falling off in E_R with excessive back pressure.

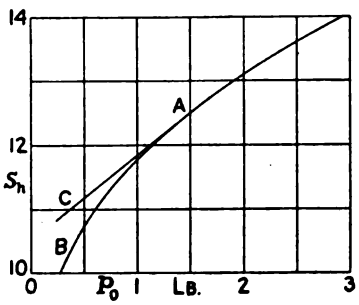

FIG. 344. — Variation of Steam Rate with Exhaust Pressure.

The last test group in Table 23, No. 36, is an interesting example of a turbine with more vacuum than it can effectively utilize, at least at full load. With half load, less steam admitted, and lower stage pressures, the last stages have less excess of energy and the general

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efficiency ratio is better than at full load: this accounts for the apparent departure from the usual relation exhibited in Fig. 340. Test 31 is another case of excessive vacuum with consequent low value of E_R. Beside lowering the relative efficiency, this condition may be carried so far as to pass the limit of increase of absolute efficiency E, in a manner analogous to that set forth for the engine in Fig. 128.

(*i*) INFLUENCE OF SPEED. — Continuing the discussion in § 46 (*g*), we now consider the question of how working efficiency is affected by change from normal action, which is shown as case I in Fig. 345, where

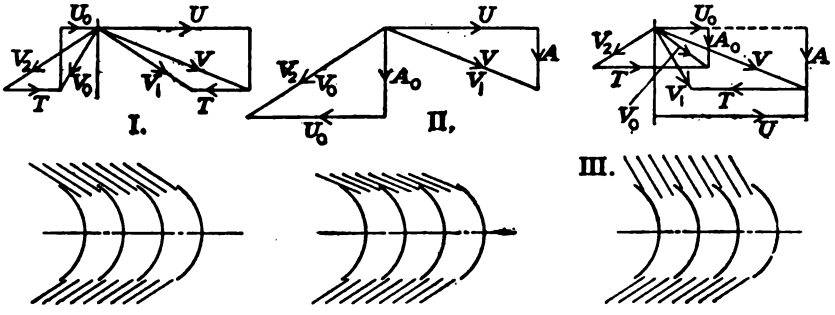

FIG. 345. — Various Speeds with the Same Vanes.

the short parallel lines above and below the vane profiles indicate relative directions of steam flow. At II, as the extreme of underspeeding, the vanes are held fixed; then with the assumption of no bucket loss, represented by making V_2 equal V_1, no kinetic energy would be abstracted from the jet — but compare § 47 (*n*), under *Observations of Pressure*. At III the vane speed T is so large that absolute exit velocity V_0 is inclined in the same general direction as jet velocity V; further, account is taken of bucket loss, in that V_2 is made less than V_1. From the profile diagram it appears that poor entrance conditions, with the jet striking the approaching, convex side of the vane, will tend to augment the secondary losses.

Steam turbines are not likely to be run above their designed speed; but relative overspeeding, essentially equivalent to case III of Fig. 345, occurs when the jet velocity is much diminished at light loads, in ordinary constant-speed machines. Underspeeding, with resulting loss in the form of residual kinetic in the jet at exit from the vanes, is a trouble belonging rather to marine service.

(*j*) SPEED AND ENERGY LOSSES. — By experiment it has been found that the friction of steam on metal surfaces varies about as the square of the relative velocity, ranging from the 1.8 to the 2.0 power. For

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rotor friction, this resistance is multiplied by speed to get power absorbed, so that the latter varies nearly as the cube of revolutions per minute. This fact tends to lower the speed of maximum efficiency from the ideal value deduced in § 46 (g), most strikingly in a noncondensing impulse turbine with a number of wheels and consequently a large amount of moving surface.

The effect of speed change upon energy losses is well illustrated by the experiments represented in Fig. 346. This simple turbine was especially adapted to feeling and showing the effect of speed alone, as an independent variable; in a more complex machine, change of speed

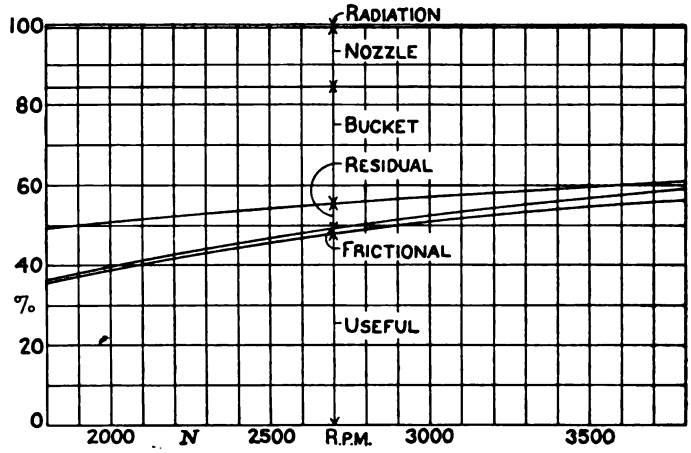

FIG. 346. — Results from a Large Single-stage One-impulse Riedler-Stumpf Turbine No. 17 in Table 20. Experiments and deductions by F. Rötscher, *Zeit. Ver. deutsch. Ing.*, 1907, Vol. 51 I. One wheel 78.8 in. in diameter, normal speed 3000 r.p.m. Initial pressure 200 lb. abs., superheat 125 deg. fahr., exhaust pressure 1.3 lb. abs.

will involve some change in other conditions, such as stage pressures. Useful output was measured as shaft horse-power, nozzle and frictional losses were found by separate experiments, and the remainder was apportioned among bucket and residual-velocity losses by inference. The best relative efficiency attained in this series of tests, even at 3800 r.p.m., is quite a little below that in test 17.4, Table 20, where the speed was only 3000 r.p.m. but the exhaust pressure was higher. Naturally, a much abbreviated turbine such as this will not be capable of utilizing high vacuum (or low exhaust pressure) to good effect.

(*k*) PERFORMANCE BY STAGES. — A typical temperature-entropy diagram for the steam turbine is given in Fig. 347, using an example with comparatively few stages. The whole available energy, area

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$A_1B_1CE_4$, is divided into four equal parts by the lines A_2B_2, A_3B_3, A_4B_4. Let the relative efficiency in each stage be 0.65, and in stage 1 draw line G_1H_1 to divide area $A_1B_1C_1E_1$ in the ratio 0.65 to 0.35: then $A_1G_1H_1E_1$ represents the useful energy given to the wheel and $G_1B_1C_1H_1$ is the total waste. Assuming zero radiation, the latter is all supposed to be returned to the steam at the pressure of line A_2B_2, increasing its entropy by the amount C_1B_2. The available energy for the second stage is now $A_2B_2C_2E_2$, and G_2H_2 is drawn so as to cut off 0.65 of this in $A_2G_2H_2E_2$. The final result of applying the efficiency 0.65 in every stage is to make the area between A_1E_4 and the GH lines equal to 0.685 of $A_1B_1CE_4$: and the excess of this ratio over the mean of the partial ratios shows the gain resulting from the return of energy not utilized in the early stages.

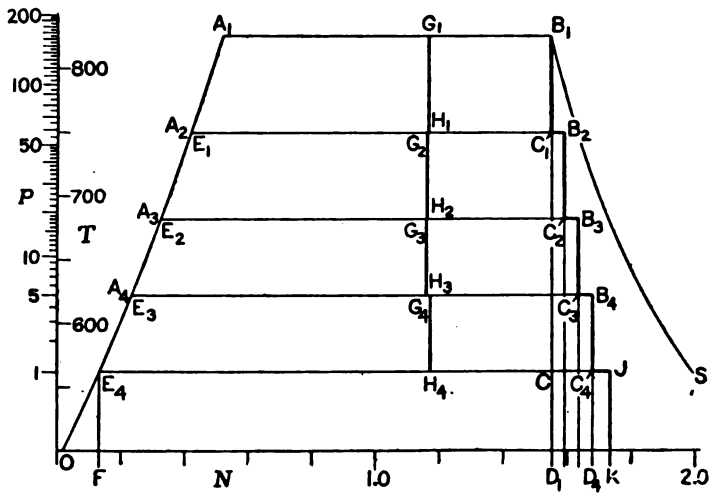

FIG. 347. — Temperature-entropy Diagram for a Four-stage Turbine: p_1 = 165 lb. abs., steam initially dry-saturated, p_0 = 1 lb. abs.

It must be clearly understood that the GH lines in Fig. 347 are not lines of operation, or do not represent any operation within the turbine: each of them simply sets off an area equivalent to that part of the ideally available energy which, as the result of very complex secondary actions, is not usefully applied in driving the rotor.

(*l*) THE MOLLIER DIAGRAM. — The heat-entropy or Mollier diagram — see general description in § 18, page 138 — is by far the best scheme of representation for turbine performance. An illustrative example, worked out with data from an actual test, is diagrammed in Fig. 348 and given numerically in Table 24. The turbine has four two-

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impulse stages and is essentially similar to the Curtis machine in Fig. 23; but it is much smaller and is horizontal instead of vertical. A special feature is the control (by hand) of the number of nozzles open for passage of steam in the three diaphragms between wheels.

In Fig. 348, line 1A represents adiabatic expansion in the first nozzles and A2 shows the return at p_2 of all the energy not usefully applied. Horizontal line 2K divides the ideal heat drop or available energy 1A into realized output 1K and return KA. The succeeding

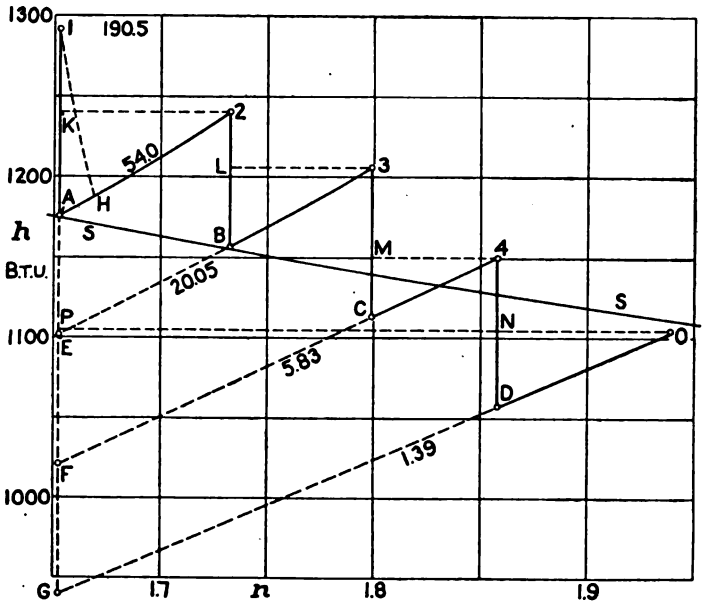

FIG. 348. — Mollier Diagram from 450 Kw. Four-stage Schulz Turbine, Test 34.2 in Table 20. Steam per brake horse-power-hour, S_h = 13.77 lb.; work output per pound of steam, W = 184.8 B.t.u. Other data, with calculated quantities in Table 24.

stages are similarly represented by 2B3L, 3C4M, and 4DON. In stage 1, dotted line 1H shows the probable actual expansion in the nozzle, with some loss of energy in friction, etc.; it is drawn for 10 per cent nozzle loss, or the vertical height from A to H is one-tenth of A1. For the whole expansion, 1G, GO, and OP show what would be an equivalent effect in ideal jet formation, heat return, and resultant efficiency: heat 1P is the useful output and 1P/1G the efficiency ratio.

The values in Table 24 are all calculated from the steam tables, but in practical use of the Mollier diagram (in service form) they would be found graphically. A highly essential feature of this test is the fact

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that the steam is superheated at the points 2, 3, and 4, or at entrance to the second, third, and fourth sets of nozzles. If these stage points lay below the saturation line SS, their positions would be indeterminate: that is, the heat content of the steam at entrance to the stages would be unknown, and could be found only by the use of some sort of steam calorimeter. No determinations of this sort for wet steam in turbine stages have as yet been made, or made public.

TABLE 24. OBSERVED AND CALCULATED QUANTITIES FOR TURBINE TEST REPRESENTED IN FIG. 348.

Quantity.	Whole cycle.	Stage. 1	2	3	4
1. Pressure.....p_1...	1 190.5	1 190.5	2 54.0	3 20.05	4 5.83
2. Temperature...t_1....	1 543.0	1 543.0	2 415.4	3 332.6	4 206.6
3. Sat. temp.....t_s....	1 377.8	1 377.8	2 285.9	3 228.1	4 168.8
4. Superheat.....s....	1 165.2	1 165.2	2 129.5	3 104.5	4 37.8
5. Total heat.....h_1....	1 1291.9	1 1291.9	2 1240.7	3 1205.8	4 1150.1
6. Entropy.....n....	1 1.6525	1 1.6525	2 1.7326	3 1.7991	4 1.8580
7. Actual.....h_2...	0 1104.8	K 1240.7	L 1205.8	M 1150.1	N 1104.8
8. Output.....W...	1P 187.1	1K 51.2	2L 34.9	3M 55.7	4N 45.3
9. Ideal.....h_2....	G 940.0	A 1175.2	B 1156.7	C 1113.1	D 1057.7
10. Output.....W_R..	1G 351.9	1A 116.7	2B 84.0	3C 92.7	4D 92.4
11. Rel. eff.....E_R ..	0.532	0.439	0.416	0.601	0.490

Small figures and letters at left edge of each column refer to Fig. 348.

The calculations for Table 24 involve no new points, unless, perhaps, in the idea of measuring actual heat content at the common state of exit from one stage and entrance to the next; and they are illustrated in the example which follows. Terminal point O is in the wet-steam region, and must be located indirectly. A note under Fig. 348 states that the output per pound of steam, based on brake horse-power, is 184.8 B.t.u. Work used up in friction of wheels in steam is properly a part of the heat returned; but purely mechanical friction is altogether outside of steam action. As very moderate allowance for bearing friction and governor resistance, $1\frac{1}{4}$ per cent is added to 184.8, making the net steam-work output 187.1 B.t.u. Measured down from 1, this locates the division point P on 1G, and the intersection of constant-heat line PO with constant pressure line GO fixes the terminal point O, for the whole cycle and for stage 4. A minor inaccuracy in the general scheme of the diagram is the disregard of radiation; but the amount of heat involved in this action is relatively very small.

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EXAMPLE 51. — Make calculations for the third stage in Fig. 348 and Table 24, and get the abscissa of terminal point O.

The first operation is carried out as in Example 49. With pressure p_1 = 20.05 lb. abs. and temperature t_1 = 332.6 deg., total heat h_1 is read from Table VII as 1205.8 B.t.u. and entropy n from Table VIII as 1.7991; the latter locates line 3C, also, at pressure p_2 = 5.83 lb., fixes moisture m_2 as 0.0198 and total heat h_2 as 1113.1. For the same pressure 5.83 lb. and the observed temperature 206.6 deg. (in the column for stage 4), total heat by Table VII is 1150.1 B.t.u. Now actual output is W = 1205.8 − 1150.1 = 55.7, and ideal output is W_R = 1205.8 − 1113.1 = 92.7; then relative efficiency E_R is 55.7 ÷ 92.7 = 0.601.

To get point O, we have that final heat content h_0 = initial heat 1291.9 − output 187.1 = 1104.8 B.t.u. At the exhaust pressure 1.39 lb., total heat H = 1109.3 and latent heat r = 1028.4; then a heat deficiency of the value 1109.3 − 1104.8 = 4.5 corresponds with the moisture fraction m = 4.5 ÷ 1028 = 0.0044. With N = 1.9481 and b = 1.7957, the final entropy is n_0 = 1.9481 − (0.0044 × 1.7957) = 1.9481 − 0.0079 = 1.9402.

(*m*) THE COMBINED UNIT. — The combined working of a high-pressure engine and a low-pressure turbine is well represented by the tests set forth in Table 25. In a few points this workup differs from that compressed into Table 20. The same idea is followed of changing from generator output to shaft power, even though the electrical efficiency of the generator must be assumed; but here the real turbine output is left in kilowatt measure, in lines 5 to 7. All work and energy quantities are related to one pound of steam entering the engine, hence to less than one pound of that entering the turbine: the exact relative weight of the turbine steam is given in line 20. This scheme makes the heat received by the turbine, line 36, exactly comparable with that rejected by the engine, line 35; the difference (around 40 B.t.u. in the last three tests, greater in the first) is lost in radiation and in the hot water drained from the engine receiver and from the intermediate separator.

The relative efficiency of the turbine, while fair, is below that of several smaller machines which precede it in Table 20. This may be due to excessive moisture in the steam flowing from the separator to the turbine. As in all the examples worked out in this book, initial heat content is calculated for actual steam quality: that is, heat h_1 in line 36 is for a steam-and-water mixture containing the water fraction m_1 in line 21; but it is for less than one pound of that mixture, as has just been stated. The engine shows an excellent relative efficiency, due to large size and small thermal losses in the cylinders — refer to remarks under Fig. 137, page 256.

In absolute efficiency this combined unit is equaled by turbines 27, 30, 31, 40, 41, 42, and 46 in Table 20, and is surpassed only by

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TABLE 25. COMBINED WORKING OF ENGINE AND TURBINE, CAPACITY 16,000 KW., INTERBOROUGH POWER HOUSE, NEW YORK.

Original test number	44	60	54	38	
Number in Table 20	50.5	50.2	50.3	50.4	
Number in Table 13	27.2	27.3			
Diagram in Fig. 137	B	C			
POWER AND WORK OUTPUT					
1. Generator output, engine K_e	4940	6923	7820	8384	kw.
2. Generator output, turbine K_t	4426	4860	6283	7784	"
3. Electrical efficiency, engine	0.960	0.967	0.972	0.975	
4. Electrical efficiency, turbine	0.958	0.960	0.965	0.970	
5. Shaft output in kw., engine K_{se}	5150	7155	8040	8600	kw.
6. Shaft output in kw., turbine K_{st}	4620	5055	6510	8030	"
7. Shaft output in kw., unit K_s	9770	12210	14550	16630	"
8. Steam per hour, to engine	131080	159760	195745	244905	lb.
9. Steam per hour, drained out	8390	17830	19180	7425	"
10. Steam per hour, to turbine	122690	141930	176565	237480	"
11. Steam per shaft kw. per hour, engine S_{ke}	25.48	22.32	24.35	28.52	lb.
12. Steam per shaft kw. per hour, turbine S_{kt}	28.40	31.60	30.03	30.52	"
13. Steam per shaft kw. per hour, unit S_k	13.42	13.08	13.44	14.74	"
14. Output per pound of steam, engine W_e	133.9	152.8	140.2	119.7	B.t.u.
15. Output per pound of steam, turbine W_t	120.1	108.0	113.6	111.8	"
16. Output per pound of steam, unit W	254.0	260.8	253.8	231.5	"
STEAM CONDITIONS					
17. Initial press. abs. p_1	198.8	195.1	193.4	197.0	lb.
18. Initial quality m_1	0.013	0.010	0.011	0.005	
19. Intermediate pressure p_i	16.10	12.10	15.18	20.60	lb
20. Steam to turbine, weight	0.9360	0.8883	0.9020	0.9697	"
21. Steam to turbine, quality m_t	0.079	0.027	0.031	0.098	
22. Exhaust pressure p_0	0.46	0.43	0.46	0.74	lb.
WHOLE UNIT					
23. Initial heat content h_1	1187.5	1189.8	1188.8	1193.9	B.t.u.
24. Ideal final heat h_2	821.4	820.6	823.4	843.1	"
25. Exhaust-feed heat q_0	45.1	43.1	45.1	60.0	"
26. Ideal output W_R	366.1	369.2	365.4	350.8	"
27. Relative efficiency E_R	0.693	0.707	0.695	0.660	
28. Input of heat Q	1142.4	1146.7	1143.7	1133.9	B.t.u.
29. Absolute efficiency E	0.222	0.227	0.222	0.205	
30. By experimenters E'	0.211	0.218	0.212	0.195	
ENGINE ALONE					
31. Initial heat h_1	1187.5	1189.8	1188.8	1193.9	B.t.u.
32. Ideal final heat h_2	1007.4	991.8	1005.3	1027.7	"
33. Ideal output W_R	180.1	198.0	183.5	166.2	"
34. Relative efficiency E_R	0.743	0.773	0.764	0.720	
35. Heat from engine	1053.6	1037.0	1048.6	1074.2	B.t.u.
TURBINE ALONE					
36. Initial heat h_1	987.4	994.7	1010.6	1029.8	B.t.u.
37. Ideal final heat h_2	819.8	820.8	825.4	848.5	"
38. Ideal output W_R	167.6	173.9	185.2	181.3	"
39. Relative efficiency E_R	0.717	0.622	0.613	0.617	

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NOTES ON TABLE 25.

Engine 42 and 86 by 60 in. duplex compound, rated 5000 kw., No. 27 in Table 13. Turbine 5000 kw. 3-stage Curtis, No. 50 in Table 20. H. G. Stott and R. J. S. Pigott, Jour. A.S.M.E., Mar., 1910. Only primary data used in workup, as noted below; derived quantities, notably water rates, are inconsistent in original report.

NOTES ON ITEMS.

1, 2, 8, 9, 10, 17, 18, 19, 21, 22; from original tables of data.

3, 4; assumed at probable values, to change basis of efficiency to shaft output.

5, 6, 7; shaft output, or power delivered to generator, but expressed in kilowatts instead of horse-power.

9; Water drained from engine receiver and from intermediate separator; note how a low value of this quantity corresponds with a high value of moisture in steam to turbine, line 21.

11, 12, 13; these are all from total steam to engine or to plant, line 8. Rates marked "Actual" in Table 12 of original report are really dry-steam rates, with simple subtraction of entrained water.

14, 15, 16; divide 3412 by S_k to get W; see § 26 (c).

20; weight of steam going to turbine, per pound of steam to engine, or line 10 ÷ line 8.

21; this m_t is fraction of water in steam to turbine.

23, 31; for one pound of steam, at pressure in line 17 and quality in line 18.

24; calculated for adiabatic expansion to exhaust pressure p_0 in line 22.

25; heat in feed water, if at temperature corresponding to p_0.

26, 27, 28, 29; same meaning and methods as in Table 20; actual output W in line 16.

30; as derived in original report, from actual electrical output and hot-well temperature.

31, 32, 33, 34; same meaning and methods as in Table 20, actual output W_e in line 14; first two tests are in Table 13, but performance is there worked for indicated instead of shaft horse-power.

35; whole heat rejected by engine, or initial heat h_1 in line 23 or line 31 less engine output W_e in line 14.

36, 37, 38; taken from Table 20, but diminished to correspond with weight of steam entering turbine (per pound of initial steam), as given in line 20. The value of W_t in line 15 bears this same ratio to that of W in Table 20.

Nos. 36 and 43. This result is attained without superheat, and with wetter steam going to the turbine, even after the drainage of 5 to 10 per cent of water, than would go to the lower stages of the high-superheat turbines. In spite of these handicaps, the relative efficiency of the plant, line 27 of Table 25, is in the region of the best attained by the turbine alone under the most favorable conditions.

With the engine for efficient utilization of the high-pressure end of the cycle and the turbine for the low-pressure end, and with both machines designed from the beginning for combined operation, the most economical steam plant possible can undoubtedly be evolved. A relative efficiency of 75 per cent — or, with the regenerative feed-heating scheme, of 80 per cent — does not seem an extravagant prediction for big units. Whether the advantage of the piston engine over the high-pressure stages of the turbine is enough to overbalance its higher first cost, carrying thermal gain over into the column of resultant gain in total cost, is another question, not to be gone into here.

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(*n*) ESTIMATE OF TURBINE LOSSES. — A review of the column for relative efficiency E_{Rh} in Table 20, with Fig. 339, will show that the combined losses in turbines from 1000 kw. rating upward ranges from 0.40 to 0.30 of the available energy. Of this, about one-sixth is probably nozzle loss and another sixth may be lost in residual velocity and rotor friction, leaving the remaining two-thirds as bucket loss (including the effect of leakage). This statement is, of course, a mere approximate summary of empirical knowledge. To rationalize the waste actions in the turbine is at least as difficult a problem as to rationalize the thermal action of the cylinder walls in the engine — and is as far from complete solution. Much effort has been expended in the attempt to develop a mathematical theory of steam friction: it is well established, as stated in Arts. (*f*) and (*j*), that surface friction is about proportional to steam density and to the square of velocity; but even if these were rigid laws, their application to conditions of rapidly changing density and velocity would be exceedingly complicated. This task is the less worth while because of the large relative magnitude of the wholly indeterminable effect of steam shock, eddy currents, etc.; under "steam shock" may be included such variations of pressure within vane channels as are described in § 47 (*n*).

There are several good reasons why a formulation of empirical information on the detail of steam action in the turbine — analogous to that made for the engine in §§ 22 to 25, Chapter V — should not be attempted here. The body of information is as yet but fragmentary, although a good deal has been published in the last few years, especially from German sources;* the subject is decidedly one for the specialist in turbine design rather than for the general power engineer; and it belongs properly to a large and full treatise such as that of Stodola, not to a textbook for engineering students.

In brief summary of the last two sections of this chapter, it may be stated that the ideal jet as worked out in § 16 will be modified in the actual nozzle to the extent of 5 to 10 per cent; while the "theoretical" bucket action in § 46, in respect to the effective performance of work, fails of realization by as much as 25 to 40 per cent. The larger the vanes and the easier their curves, the smaller will be the losses, both by surface friction and by secondary dynamic actions.

* Statement written in summer of 1911.

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CHAPTER X

DESIGN AND CONSTRUCTION OF THE TURBINE

§ 49. Design for Steam Action

(*a*) DIMENSIONS OF STEAM CHANNEL. — In laying out an engine which shall develop a certain power, the principal step is to determine proper sizes for the cylinders, and the machine is then built up around these volumes. The analogous fundamental determination in the design of a turbine is that of the channel (through nozzles and buckets) along which the working current will flow. The essential requirements are that this channel be of the right size to carry the necessary amount of steam, and that it be properly graded in sectional area so as to permit and produce correct expansion of the steam. We shall here go no farther than this primary design for steam action.

Having given initial pressure and state of steam and exhaust pressure, the ideal output per pound is to be calculated for the Rankine cycle as in § 15 (*d*), or read from a Mollier diagram: the method of calculation is more clearly illustrated in Example 49, page 469, than in Example 15, page 105. Then the assumption of a probable relative efficiency will lead to the actual work per pound of steam; and from this, with the desired power of the turbine, the total steam per hour, minute, or second is readily found. The simplest case is that of the single-stage, one-impulse turbine, which is little more than a nozzle problem.

EXAMPLE 52. — Assuming ideal steam action, find (A) the combined nozzle area for a 300 brake horse-power turbine, with dry saturated steam at 150 lb. abs. and an exhaust pressure of 1.5 lb. abs. Then see how the capacity of this turbine will be affected by (B) change to steam with 10 per cent of moisture and (C) to steam with 150 deg. of superheat.

First get the ideal output in all three cases, as follows:

Case	A	B	C
1. Initial total heat, h_1	1193.8	1107.4	1277.3 B.t.u.
2. Entropy, adiabatic expansion, n . .	1.5697	1.4642	1.6640
3. Dry-steam entropy at p_0, or N_2 . .	1.9417	1.9417	1.9417
4. Entropy shortage, $N_2 - n = \Delta n$.	0.3720	0.4775	0.2777

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Case	A	B	C
5. Entropy of vaporization at p_0, or b_2 .	1.7846	1.7846	1.7846
6. Final moisture fraction, m_2 . . .	0.2085	0.2674	0.1555
7. Final total heat, h_2	896.5	835.8	950.8 B.t.u.
8. Ideal output, W_R	297.3	271.6	326.5 "

9. Now assume a relative efficiency of 0.55; then the output per pound of steam in case A will be $W = 297.3 \times 0.55 = 163.5$ B.t.u.
10. The steam per horse-power-hour will be $S_h = 2545 \div 163.5 = 15.56$ lb. Then for 300 h.p. the total steam will be 4670 lb. per hour or 1.298 lb. per sec., say 1.3 lb.

Next, calculate the throat dimensions of a unit ideal jet for each initial condition, being guided by Fig. 65 in choosing the throat pressure.

Case	A	B	C
11. Pressure at throat of jet, p_t . . .	87.0	88.0	86.0 lb.
12. Dry-steam entropy, N_t	1.6135	1.6126	1.6144
13. Steam quality at p_t, m or s . . .	0.0380	0.1291	53.7
14. Heat content at p_t, or h_t	1149.7	1068.4	1223.6 B.t.u.
15. Energy in jet, $h_1 - h_t = W_t$ . . .	44.1	39.0	53.7 "
16. Velocity of jet, V_t, by Eq. (104) . .	1484	1410	1640 f.p.s.
17. Specific volume, v_t	4.862	4.355	5.609 cu. ft.
18. Area of throat, a_0	0.472	0.445	0.492 sq. in.
19. Napier divisor, $D = a_0 p_1$	70.8	66.8	73.8

20. Since the turbine requires 1.3 lb. of steam per sec. in case A, and the throat area of the unit jet is 0.472 sq. in. (this can be read from Table 7 also), the total least cross area of the nozzles must be $0.472 \times 1.3 = 0.613$ sq. in. The ideal rate of energy availability is $1.3 \times 297.3 = 386$ B.t.u. per sec.
21. In case B the area just fixed will discharge $0.613 \div 0.445 = 1.378$ lb. of steam per second, of which the available energy will be $1.378 \times 271.6 = 374$ B.t.u. Relative efficiency will probably be less with wet steam, so that the power of the turbine will be about 5 per cent less than in case A.
22. In case C, the steam flow will be $0.613 \div 0.492 = 1.2460$ lb., which will have an available energy of $1.246 \times 326.5 = 407$ B.t.u. It appears therefore that the use of superheated steam will increase the power of a turbine with given nozzles.

This example carries forward the investigation of Fig. 65. There it was found that rate of flow increased with wet steam, decreased with superheat, for a given orifice area: now we see that these effects are overbalanced by changes of available energy (per pound) with initial steam condition. We have not taken into account the coefficient of discharge, as discussed in § 47; this would slightly modify the absolute rates of steam flow, and might make a small change in their relative size.

(*b*) THE MULTISTAGE TURBINE. — Suppose that a turbine has a number of equal stages, with the same kinetic energy and velocity of

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the steam at the mouth of each successive set of nozzles. Then the area of channel in each stage, in nozzles and through vanes, will be directly proportional to the specific volume of the steam, at the condition in which it leaves the nozzles. This volume is shown by the respective lengths G_1H_1, G_2H_2, etc., in Fig. 349; where the stepped curve BC corresponds with the outline $B_1C_1B_2{}^{***}C_4$ in Fig. 347, and BD with the plain adiabatic B_1C. For equal stages, areas ABH_1G_1, $G_1H_1H_2G_2$, etc., in Fig. 349, will be equal to each other; and, as just stated, the nozzle areas will bear the same ratio to the volumes G_1H_1, G_2H_2, etc. With unequal stages, this ratio will change; if the energy

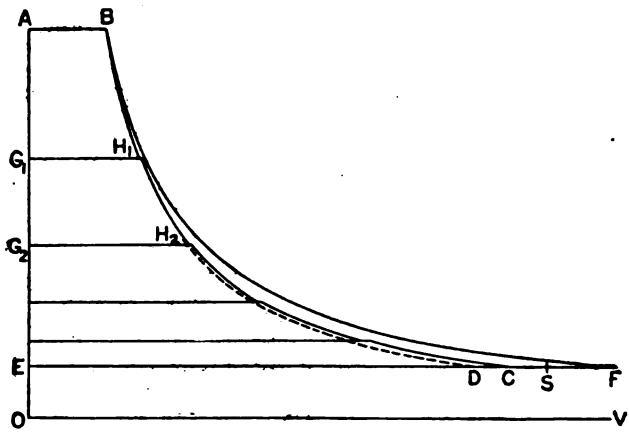

FIG. 349. — Pressure-volume Diagram for a Multistage Turbine.

and steam velocity of the stages are increased, the cross area required for steam flow will be relatively less. The successive enlargements of rotor diameter in a reaction turbine like Fig. 25 are in accord with this principle.

Quantitatively dependent upon entropy relations, the pressure-volume diagram as in Fig. 349 is of little or no use for primary determinations, whether of stage pressures or of jet dimensions. Its purpose is wholly illustrative: and in pursuance of that idea, the line of constant total heat is drawn in curve BF, while S is a point on the saturation curve from B.

To divide the original ideally available area $A_1B_1CE_4$, Fig. 347, into equal parts (or in some other definite manner) is a comparatively easy task; but the return of unused heat from the higher stages disturbs the proportions of such an initial division. The stage areas $A_1B_1C_1E_1$, $A_2B_2C_2E_2$, etc., can be equalized only by a very troublesome cut-and-try adjustment of the heights of lines A_2B_2, A_3B_3, etc., or of the stage

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pressures. The Mollier or heat-entropy diagram, already exemplified in Fig. 348, is far more useful and convenient for this kind of determination than is the plain temperature-entropy diagram.

(*c*) USE OF THE MOLLIER DIAGRAM. — Drawn for illustrative purposes, Fig. 350 is made much clearer and more open than the service diagram: only the lines of constant pressure are put in (with one quality curve in the saturation line SS); and these are spaced for pressures corresponding to each ten degrees of saturation temperature, as in Fig. 72. With the steam limits named, line AB represents the operation of

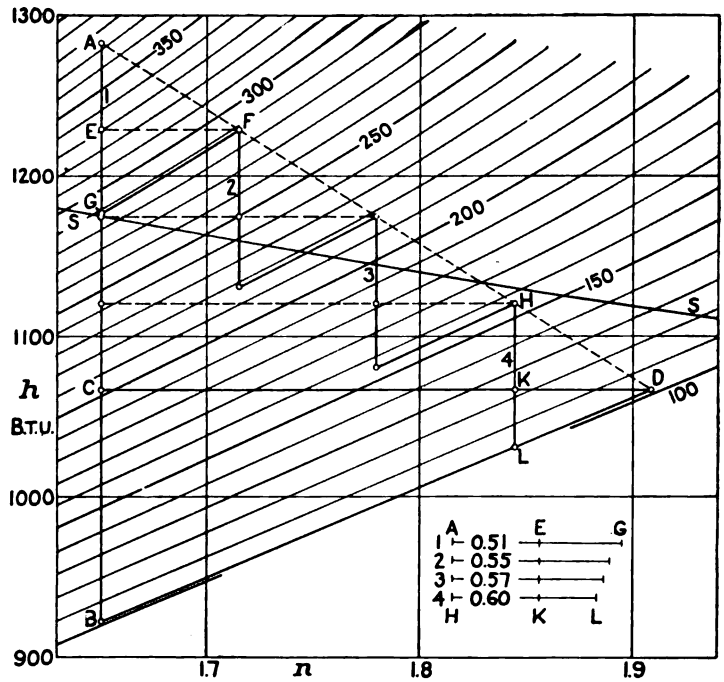

FIG. 350. — Diagram for Four-stage Turbine, working from 180 lb. abs. and 150 deg. of superheat to 1 lb. abs.

ideal jet formation and application. The relative efficiency being taken as 0.60, AC is made equal to 0.6 of AB; then intersection of the horizontal or constant-heat line CD with the exhaust-pressure line BD locates the final state point at D.

As has been indicated, the solution of the problem of dividing the whole work performance in a desired manner among the stages consists in fixing the intermediate stage pressures. Obviously, division of the length AB into four equal parts would be equivalent to a similar

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division of the area $A_1B_1CE_4$ in Fig. 347; and locations with reference to the pressure and temperature lines of Fig. 350 could be transferred directly to the temperature axis of Fig. 347. This scheme of division will, however, give a larger energy value to the lower stages, as has already been pointed out.

If equality of the stage energies, represented in Fig. 350 by the vertical lines from AG or 1 to HL or 4, be desired, the problem has no simply definite solution. The individual stage diagrams, like AGFE and HLDK are generally similar to the overall diagram ABCD; but even with a fairly simple expression for the constant-heat lines in terms of the coördinates of the diagram, relations among the stage quantities would be rather complex. Lacking such a foundation for mathematical determination, it is necessary to fall back on the cut-and-try method; although with the advantage that it is easier to handle heat and energy as linear quantities, than as areas in the original temperature-entropy diagram. Before going farther with this particular problem of stage equalization, we will consider a purely graphical scheme, which is outlined in Fig. 350.

(*d*) THE STAGE-POINT LOCUS. — In that diagram, the useful output AC is spaced off into four equal parts, implying the same net work performance in all the stages. From the division points horizontal (dotted) lines are extended toward the right, until they meet the stage-point locus AD. The example shows the result of using a straight line for this locus: in the clear space at the bottom of the figure are laid out the lengths of the stage energies AG to HL, divided into output AE to HK and returned heat EG to KL. The numbers in the lines are relative efficiencies, from AE/AG = 0.51 to HK/HL = 0.60. Some such increase of relative efficiency with fall of pressure is highly probable — compare statements in § 48 (*f*) and showing of Fig. 348 — even though the exact results in the diagram are not likely to be realized.

In further illustration of the scheme just described, Fig. 351 shows the partition of the same total output into ten stages, of which the last four are together equal to the first six. The upper half of AC is divided into six twelfths, the lower half into four eighths, and the stage points are again determined by a straight-line locus AD. In the group of horizontal parallel lines representing the stage energies, lengths from the main diagram are doubled.

The simplest way to modify the stage quantities, while preserving a reasonable and consistent relation among them, is to substitute a slightly curved line for the straight locus AD: this will be concave upward, or will sag below AD, and will be used with the same skeleton of horizontals, from the same division points on AC. The shape of

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curve that would, for instance, equalize the stage or nozzle energies (like AG to HL, Fig. 350) cannot be definitely formulated, since it will change in form with the positions of A and D. Approximation by

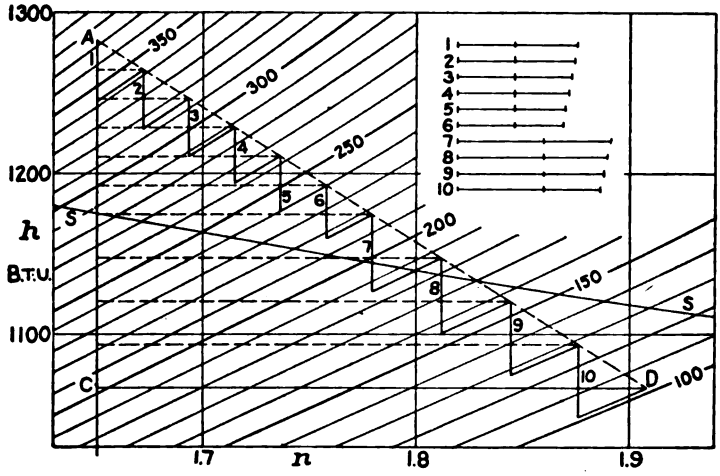

FIG. 351. — Diagram for Ten Stages.

trial is still necessary, but the operation has now as little as possible of the random element.

(*e*) FORM OF THE MOLLIER DIAGRAM. — If the constant-pressure lines were parallel, the scheme of Fig. 350 would make all the stages alike, establishing a constant ratio of output to jet energy. Since the entropy gained with a certain amount of heat is less as the temperature is higher, the lines of constant pressure rise with increasing steepness. This makes them diverge from left toward right; and the growth, with higher steam quality, of the vertical distance between any pair, represents the increase in energy available for conversion into work. The same increase is shown in Example 52; but the diagram, once drawn with sufficient accuracy, greatly surpasses numerical calculation in convenience.

(*f*) PROPORTIONS OF TURBINE STAGES. — One or the other of the ideas just set forth — either to equalize or in some other ratio to relate the jet energies, or to aim at a similar relation among net outputs — will naturally govern turbine design. As between the two schemes, there is probably little to choose in the matter of overall efficiency. The stage pressures having been decided upon, nozzle areas will be calculated by the methods of Example 52, with the help of the volume lines when using a Mollier diagram.

Only with a comparatively small number of stages in the turbine

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will divergent nozzles be required. The minimum number for straight nozzles may be determined, roughly, by applying the relation expressed in Eq. (144), page 287, letting R and r stand for ratios of pressures rather than of volumes. This relation $R = r^n$ is strictly true only with the expansion curve $pv = C$; but as appears from Fig. 349, the resultant curve BC does not depart very greatly from the equilateral hyperbola. Suppose that in a turbine with expansion from 150 lb. to 1 lb. abs. the number of stages is to be such that in each the pressure ratio (of high to low pressure) shall not exceed 1.7, the reciprocal of 0.58. Then we have $R = 150$, $r = 1.7$, whence $n = \log 150 \div \log 1.7 = 2.176 \div 0.230 = 9.5$. Since it is better to err on the side of deficiency of divergence — see § 47 (f) and last paragraph of (m) — we may have as few as eight equal stages without using divergent nozzles.

As to the control of stage pressure, in design or in operation, it will be raised by decreasing the opening through succeeding nozzles, lowered by increasing that opening. This is closely analogous to the effect of shorter and longer cut-off upon a preceding receiver pressure in a multiple-expansion engine. Generally the nozzle openings, after the first, are fixed and unchangeable; but sometimes, as in the Schulz turbine of which a test is diagrammed in Fig. 348, there is control (by hand) of the lower sets of nozzles.

To consider a different type of staging, suppose that a group of stages has a uniform channel area running through it, as is true of several groups in Fig. 25. With fall of pressure and increase of specific volume, steam velocity must be greater for equal flow rate. The stage pressures will therefore adjust themselves so as to give smaller energy values to the upper stages, larger values to the lower stages. In other words, pressure will be backed up throughout the group, its gradient running higher than for equal stages. Obviously, such interrelation of stages will be a component in the control of stage pressure as described in the last paragraph.

(g) Proportions of Vane Channels. — The sectional profiles of vanes or blades, especially the edge angles, will be governed by velocity diagrams as laid out in § 46 (d), (f), (j), and (l), with modification of impulse vanes by the considerations set forth in Fig. 337. As to effective width of channel, the relations in § 46 (k) are very little affected by the secondary departures from ideal action. In impulse turbines, the depth of channel or length of vanes is made equal to or a little greater than the diameter of nozzle mouth: with several velocity stages, of the Curtis type, the vane lengths are progressively increased, as noted in § 46 (n) and shown in Fig. 360. In the reaction turbine, length of vane is one of the main dimensions of the cross area which determines flow.

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§ 50. Various Forms of the Turbine

(*a*) TYPES OF STEAM ACTION. — Concerning this matter, little or nothing is to be added to the classification on page 32; a notable fact is the strong tendency existing toward a combination of the Curtis and Parsons types, as exemplified in Fig. 26 and in Fig. 357.

The condensed body of descriptive matter now to be given is in continuation of and in addition to the general description in § 4, Chapter I.

(*b*) DIRECTIONS OF STEAM FLOW. — All the turbines illustrated in § 4 have what is called axial flow, through nozzles and vanes as well as in general direction. A glance at flow conditions, with the more definite showing of Fig. 323, makes apparent the fact that the steam path through a series of stages is really helical in form. With partial peripheral admission, Figs. 19 and 22, there must be considerable sidewise spreading or circumferential flow of the steam current in the wheel chambers.

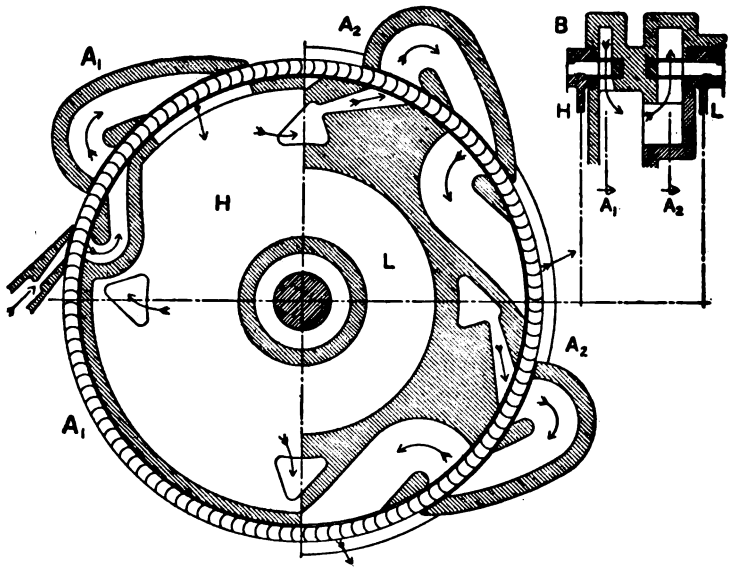

FIG. 352. — The Elektra Turbine, Radial-flow, Two-stage, Three-impulse: much structural detail omitted.

Figure 352 is an example of radial flow, alternately inward and outward through one ring of vanes. This design has the merits of compactness and, for a several-impulse type with return to the same vanes, of definite guidance of the steam current after its first velocity stage:

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but the wide and long guide channels seem to favor the formation of eddy currents. Some early Parsons turbines, in the experimental stage, had radial flow; and several recent designs of small reaction turbines show a return to this scheme, largely for the sake of compactness.

The method of tangential flow is illustrated in Figs. 353 and 354, the first with buckets of the Pelton form, adopted from the impulse type of water wheel. The use of two parallel wheels or bucket rows in one stage (at the left side of Fig. 353) is only a tentative suggestion, not carried out in practice because the steam currents would interfere in escaping from the buckets and flowing to the next nozzles.

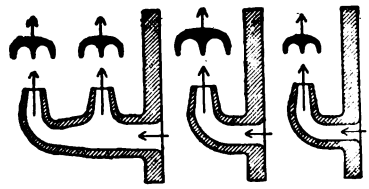

FIG. 353. — Elementary Sketch of the Kerr Turbine.

A full development of the idea of semicylindrical buckets, milled from the solid metal of the wheel rim, is shown in Fig. 354: the milling cutter

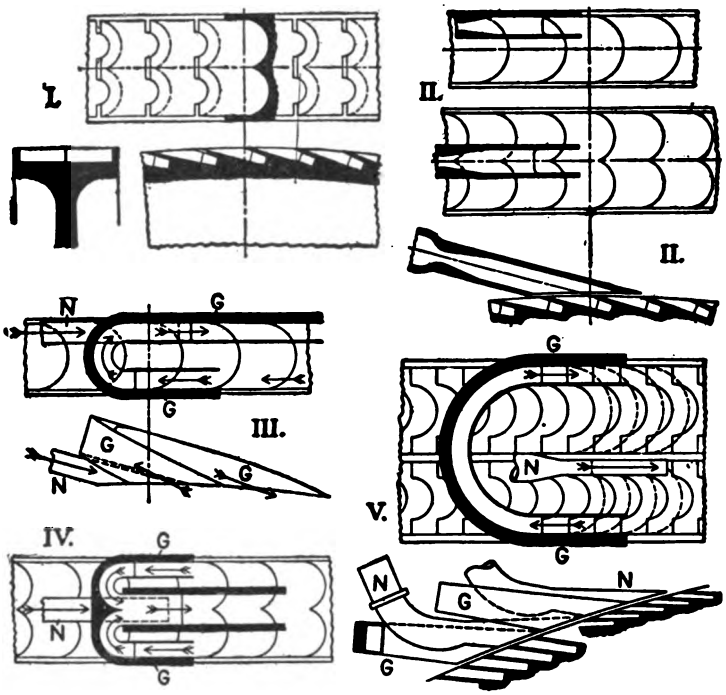

FIG. 354. — Working Elements of the Riedler-Stumpf Turbine.

is of the same form that would be used for making a straight T slot, and is so held as to give the plane of the bucket an angle of 15° or 16° with

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a plane tangent to the rim. Nozzle positions with single and double rows of buckets are indicated at II, where the second sketch represents the single-impulse arrangement of the large one-stage turbine listed as No. 17 in Table 20. The other drawings illustrate schemes of velocity staging, with peculiarly curved return channels or guide passages. At III we have simple return to the same single row of buckets: Fig. 312 shows the need for different angles (with wheel rim) at entrance to and at exit from the guide channel G. Drawing IV shows a similar return of the split current discharged from the first velocity stage on a double-row wheel. The scheme at V, with separate, larger buckets for the second velocity stage, permits closer spacing of the buckets along the circumference of the casing.

The manufacture of this turbine, never carried very far, has been given up (by the Allgemeine Elektricitäts Gesellschaft), in favor of designs with ordinary radial vanes. Several small turbines built in this country, as the Terry and Sturtevant, are similar in principle to scheme III in Fig. 354, but for return channels have fixed buckets in the casing similar to those in the wheel — see the paper, "Small Steam Turbines," by Mr. G. A. Orrok, Trans. A.S.M.E., 1909, Vol. 31, 263, reprinted in *Power* for May 11, 1909.

(*c*) VARIATIONS OF IMPULSE TURBINES. — Among this class there is scarcely any essential departure from the type of arrangement shown in Figs. 20 and 23, with the wheels in chambers separated by diaphragms. Horizontal machines have these partition discs split on a diameter, each half being fast in its part of the casing; but the Curtis turbine has solid diaphragms, which must be put in place when the wheels are being assembled on the shaft, then handled with the rotor when the machine is put together.

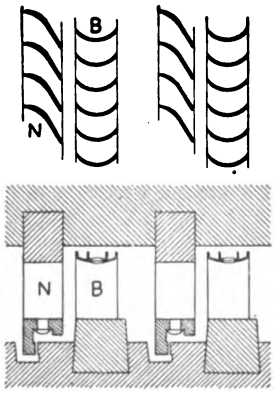

FIG. 355. — Impulse Stages, Melms Pfenninger Turbine.

The one radical departure from "normal" construction is to mount impulse vanes on a drum (as in the reaction turbine) and have the fixed nozzle rings make a running fit (as close as practicable) with the rotor. The sketch in Fig. 355, which gives no detail of the blade fastenings but merely indicates their overall bulk, shows two out of a considerable group of single-impulse stages at the high-pressure end of the Melms Pfenninger mixed-type turbine; the rest of the machine is of Parsons type, with reaction vanes on the same drum The plane "joint" between nozzle-vane ring and rotor runs with a clearance of about 0.01 in.

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A similar group of single-impulse stages on a broad wheel or short drum forms the low-pressure end of the Oerlikon-Rateau turbine; and in the marine Curtis turbine — see Art. (*g*) — the last three or four 3-impulse stages are thus mounted on one wheel. With the latter type of staging only the nozzle rings, not the intermediate guides, need run close with the rotor.

(*d*) VARIATIONS OF REACTION TURBINES. — Starting with the typical Parsons arrangement shown in Fig. 25, the first important change has been in the balance discs or pistons: Fig. 356 shows the quite largely

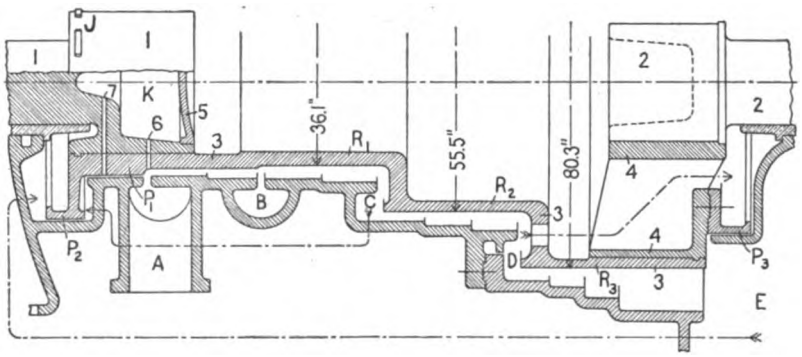

FIG. 356. — Section of Rotor and Casing, 8000 Kw. Tosi-Parsons Turbine, to illustrate Fullager system of balancing and construction of rotor. STODOLA IV, 450; *Power*, Oct. 13, 1908. Scale 1 to 42.

used Fullager system. Pistons P_1 and P_2 remain as before, but the large P_3 of Fig. 25 is replaced by the smaller P_3 in Fig. 356, at the other end of the rotor. The net area of this P_3 is equal to the annular area of step 3, at D; then the remainder of the right-hand end of the rotor is balanced by carrying the condenser pressure over to the outer face of P_2. This plan requires much less enlargement of the casing for the balance discs; and inside of one "stuffing box" (at the right) the steam pressure is not far from atmospheric, so that liability to air leakage is much diminished. In Fig. 356, regular admission is at A, and under heavy overload steam is bypassed to B.

The introduction of one or two initial, high-pressure stages of the Curtis type not only replaces the small-area reaction stages in which leakage is of serious magnitude, but also makes easy an arrangement which is more or less self-balanced — see Fig. 26. The double-flow system, represented by Fig. 357, still further facilitates the elimination of end thrust.

A good description of various forms of the Westinghouse double-flow turbine will be found in *Power* for June 16, 1908; all of them have flow

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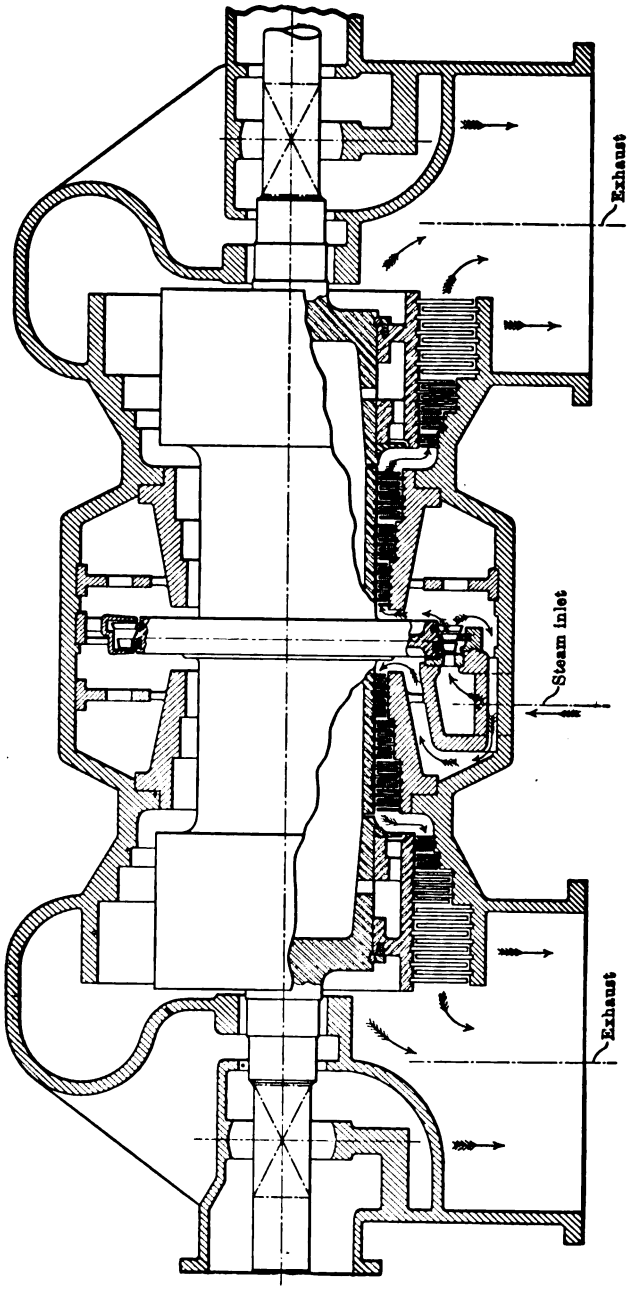

FIG. 357. — Westinghouse Double-flow Turbine, rating 10,000 kw.; No. 46 in Table 20. From Jour. A.S.M.E., Dec., 1910, page 2092.

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from the middle toward the ends, thus requiring two stuffing boxes tight against leakage of air into the condenser, just as in Fig. 25. For low-pressure service, the impulse stages are naturally omitted; then flow from the ends toward the middle, as in the Brush-Parsons turbine, No. 49 in Table 20, reduces the danger of air leakage to a minimum.

(*e*) LOW-PRESSURE AND MIXED-FLOW TURBINES. — The low-pressure turbine differs in no essential particular from the lower stages of the same type of full-range machine. If the supply of engine exhaust is irregular or intermittent, provision must be made for feeding the turbine directly from the boiler. The simplest method is to pass live steam through a reducing valve, cutting it down to the normal admission pressure of the turbine. A reaction turbine must be operated in this way, unless a special group of high-pressure stages, commonly running idle, is provided. But an impulse turbine may, more simply, have an extra set of high-pressure nozzles, of small throat area; then even though the jet from these has too high a velocity for full utilization at the established vane speed, a good part of the energy that would otherwise be wasted in pure throttling is recovered.

(*f*) THE RATEAU ACCUMULATOR. — To make the exhaust-steam turbine effective in sequence with a group of noncondensing engines somewhat irregular in operation, Professor Rateau invented his "accumulator." This is a storage vessel or tank in the steam line between engines and turbine, designed to hold heat, not steam. In some cases it has been filled with heavy slabs or discs of cast iron, but more commonly there is a tier of iron troughs, which stand full of hot water at about 212 deg. If the steam supply is less than the demand by the turbine, stored heat will be given off, with evaporation of water and some fall of temperature and pressure. An excess of exhaust will restore heat to the accumulator, raising temperature and pressure until an automatic relief valve comes into action. Complementary to the latter, there must be a bypass from the boiler, to supply reduced steam in case of a prolonged deficiency of exhaust.

(*g*) THE MARINE TURBINE. — As remarked on page 32, the most trying requirement imposed on designers of marine turbines is that of comparatively low speed, for the sake of propeller efficiency: in some recent large installations turbine speeds have been made less than 200 r.p.m., in contrast with the very common 750 r.p.m. which is a minimum for big turbogenerators. To accommodate the necessarily large number of stages, Parsons turbines are commonly made in sections; quite often, one high-pressure and two low-pressure turbines are on three shafts, the latter two with reversing rotors; while the biggest

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ships have a complete duplex arrangement, with four shafts. Under the Curtis system, enough stages can easily be put into one casing. Leading particulars from a few large ships will suffice here.

LARGE MARINE TURBINE PLANTS.

1. Steamships *Lusitania* and *Mauretania*. *Engineering*, 1907 II, 1908 I, Vols. 84, 85. Four Parsons turbines, two high-pressure, two low-pressure. High rotor drum, 96 in. diam., vanes 2.5 to 12 in. long; low drum, 140 in. diam., vanes 8 to 22 in. long. Steam pressure 210 lb. abs., speed about 190 r.p.m.; vane speed, at mid-length, 80 to 130 ft. per sec. About 60 stages in low-pressure section, twice as many in high-pressure. Combined rating, 70,000 h.p.

2. Steamships *Olympic* and *Titanic*. *Power* for July 11, 1911, etc. One low-pressure Parsons turbine, working from 9 lb. abs., in sequence with two reciprocating engines. Rotor 12 ft. diam., vanes 18 to 25.5 in. long; speed 165 r.p.m.; vane speed, at average mid-length, 120 ft. per sec. Rating, engines 30,000 h.p., turbine 16,000 h.p.

3. U. S. battleship *North Dakota*. *Power*, May 25, 1909. Two Curtis turbines, 25,000 h.p. total; steam pressure, 280 lb. abs., reduced to 75 lb. abs. in first stage; speed, 245 r.p.m. Wheels about 12 ft. diam. to mid-length of vanes, latter 1.8 to 12 in. long; vane speed about 155 ft. per sec.

(*h*) THE GEARED TURBINE. — If high machine efficiency, quiet running, and durability can be secured, the use of toothed gearing between turbine and propeller will greatly diminish the size and cost of the marine turbine, while promoting efficiency of both propeller and turbine. The Melville-McAlpine gear, designed for this service, is essentially a De Laval gear on a large scale. In *Power* for Nov. 9, 1909, will be found a description of the large experimental set of gears built for trying out that design. Other trials of the idea have been made, but the matter has hardly passed beyond the tentative stage.

In *Engineering*, 1911 I, Vol. 91, 463, is described the driving of a rolling mill by a 750 h.p. low-pressure Parsons turbine, which acts through two pairs of gears, reducing from 2000, through 375, to 70 r.p.m. See also *Power* for May 23, 1911.

§ 51. Construction of Working Parts.

(*a*) THE ROTOR of the turbine must have a proper form to carry the blades and needed strength to resist the forces which come upon it, and must be as nearly as possible in centrifugal balance, so that its rapid spinning will not cause vibrations. As to form, there are two

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distinct types: the wheel or disc of the impulse turbine and the drum or cylinder of the reaction type. In the matter of strength, the major stress is generally due to centrifugal force, although resistance to flexure by weight may become of first importance in a long horizontal turbine. The tangential driving forces, due to dynamic steam action, are relatively insignificant.

Of this, as of other working parts, only a few typical examples will be given here, with general statement of the principles involved in design and construction. For all quantitative considerations of questions of design, refer to special works, notably to Stodola's *Steam Turbines*.

(*b*) Disc Wheels. — These are commonly run at higher speeds (relative to diameter) than are drum rotors: and while the disc, properly proportioned, is far stronger than the ring against centrifugal force, such extreme speeds as are employed in the De Laval turbine tax its capabilities to the utmost. The greater strength of the disc is due to radial tension, which assists circumferential or ring tension. Since the cylindrical area across which radial stress acts grows smaller toward the axis, a disc of uniform thickness is more severely strained at the center than at the rim, and is much weakened by a central hole. The sections of De Laval wheels in Fig. 358 show how these conditions are met: by swelling the disc at the middle it is made of approximately uniform strength; in the larger sizes the shaft (in parts) is fastened to the wheel without going through it; while for the small wheel at II the weakening effect of the hole is overcome by adding a large hub. For the very complex mathematics of centrifugal stress in a disc, the reader is referred to Stodola, or to advanced works on the strength of materials; but a few general ideas are set forth in Art. (*d*).

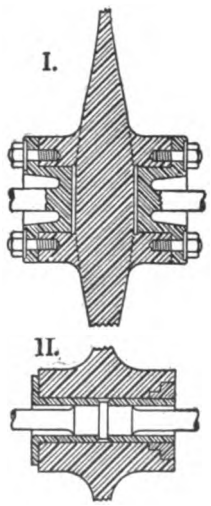

Fig. 358. — Sections of De Laval Discs. I, Rotor of larger turbines; II, Detail of shaft fastening in Fig. 18.

At the comparatively moderate speeds in multistage turbines, the problem of sustaining centrifugal force is not of extraordinary difficulty. In Chapter I, cast-steel wheels are seen in Figs. 23 and 26, discs of pressed plate in Fig. 20. Here Fig. 359 shows the present construction of Curtis wheels, with discs of rolled plate, turned thinner toward the rim; note the distance ring just outside the row of rivets near the rim, which holds the two plates at a definite distance apart. A similar two-plate construction is used for the wheels of Curtis marine turbines.

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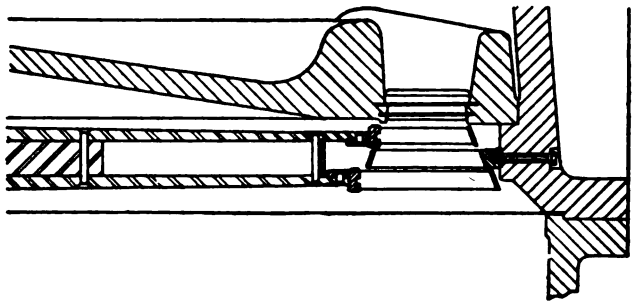

FIG. 359. — Detail of Curtis Turbine, showing construction of wheel. From current catalogue, General Electric Company.

EXAMPLE 53. — To get some idea of the magnitude of the centrifugal force on a high-speed turbine wheel, take the cases of (1) a 4-in. wheel at 30,000 r.p.m. and (2) a 30-in. wheel at 11,000 r.p.m., as named on page 24, also (3) of the 2-meter wheel at 3000 r.p.m. in § 48 (*g*). If we calculate the centripetal acceleration V^2/R or $\omega^2 R$ — compare § 31 (*b*) and (*c*), Example 35, page 300, and Example 41, page 325 — and divide this acceleration by that of gravity, we shall have the ratio of centrifugal force to weight force. The three examples evaluate as follows:

Case	1	2	3
1. Diameter, inches	4	30	78.8
2. Diameter R, feet	0.167	1.25	3.28
3. Rotary speed N, r.p.m.	30,000	11,000	3000
4. Velocity V, ft. per sec.	523	1441	1034
5. Angular speed ω, rad. per sec.	3140	1151	314
6. Centripetal acceleration a_0, ft. per sec. per sec.	1,643,300	1,656,000	324,400
7. Centrifugal ratio, $a_0 \div 32.16$	51,111	51,500	10,100

From the last line it appears that the centrifugal force on one cubic inch of steel (0.28 lb.), at the mean radii named, is about 14,000 lb. in the De Laval turbine.

(*c*) DRUM ROTORS. — In general form, Fig. 356 illustrates what may be called the regular type of construction of Parsons rotors; it is made up of shaft sections 1 and 2, shell 3, and head 4, all put together with shrunk joints: pieces 1, 2, and 3 are of forged steel, piece 4 is a steel casting. Of special features in this design, one is the joint lock at J (as at the other end also), with interrupted collars which are engaged by turning the inner piece through a small angle after it is slipped into the heated outer piece. Another is the provision for preventing the joint at the high-pressure end from possibly being loosened by excessive heating of the shell alone: space K is shut off by disc 5, and steam is

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let in through hole 6; and in order to maintain some circulation, hole 7 opens into the labyrinth packing of piston P_1 near its outer edge.

The first variation from the arrangement in Fig. 356 is to form flanged heads on the shaft sections, and bolt them fast to the shell with studs parallel to the axis; this makes the intermediate piece 4 unnecessary. One design (Willans and Robinson) has the shell forged in one piece with the shaft section at the high-pressure end. Others use a plain shell of the smallest rotor diameter, and make the enlarged sections or steps by putting on wheel-like rings, very much as in Fig. 357. Marine Parsons turbines, with high and low sections, will have cylindrical drums, or but a small change of diameter. In Fig. 357, the shaft and drum sections are steel castings, joined at the middle: in one piece with the impulse wheel and projecting equally at both sides there is a short sleeve or cylinder, which slips into the open ends of the drum sections; the latter are flanged and are held together by bolts, through the web of the wheel and parallel to the axis of rotation. A great number of illustrations are given by Stodola, in his fourth edition.

It is highly important that neither rotor nor casing of a reaction turbine shall sag, spring, or warp out of shape, whether from weight or from irregular heating, because of the close clearance at the vane ends. Skillful design, careful annealing of castings for outer casing, and accurate workmanship are required to obviate trouble in operation.

(*d*) Centrifugal Stress. — In § 33 (*e*), page 326, we have shown that the tensile stress in a ring because of centrifugal force is determined solely by linear velocity, and that it varies as the square of this V; further, that a stress of 20,000 lb. per sq. in. will result from a speed of 450 ft. per sec. Of course, the ring almost exactly represents the cylindrical shell of a drum rotor. Allowing for the added load due to the mass of the attached vanes, a speed of about 400 ft. per sec. is considered a good average limit of safe running with ordinary steel.

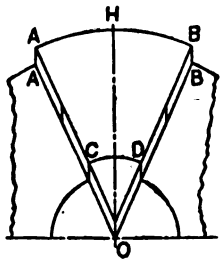

Fig. 360. — Diagram to Illustrate Stress in Disc.

To get an approximate idea of conditions in a disc, consider Fig. 360. Suppose a narrow sector to be cut loose along the lines AO and BO, and moved a little way outward in the direction OH; then all points along the radial edges OA and OB will have the same displacements from their original positions, but the movement at C, for instance, will bear a much higher ratio to the arc CD than will that at A to the arc AB. Similarly, as the disc is stretched by centrifugal force there tends to be a relatively greater deformation, hence stress, at the middle than at the

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rim. This situation is relieved by radial stretch, which eases up the pull on the central metal; but for complete equalization of stress the central part must be given the reinforcement of increased thickness.

(*e*) CENTRIFUGAL BALANCE. — The problem of bringing the center of mass to the rotation axis is first one of getting the most perfect geometrical symmetry that is attainable with machine-tool appliances, and then of trying for and correcting the residual eccentricity by special methods. To overcome the effect of the lack of absolute perfection, and to avoid the need of very precise and expensive work in balancing, De Laval devised his flexible shaft, of which the action depends upon the following principles:

If an eccentric mass is attached to a rotating shaft, and the whole mass be thought of as concentrated at its center of gravity, then the centrifugal force of this "particle" will act radially outward, tending to deflect the shaft in its own direction, and producing a pressure of shaft on bearings which constantly changes in absolute direction. This type of action has been quite fully discussed under the head of shaking force and counterbalance, in § 35.

But when we consider the mass in its actual distributed state, another force action comes into play, which reaches controlling magnitude only at very high speeds. A spinning disc tends to get into the simple, stable state of rotation about its own center of mass — this tendency being closely analogous to that of the same disc to preserve one fixed plane of rotation, or to what is called gyroscope action.

In the case of any nearly balanced rotating body, as the speed increases the eccentricity effect at first increases very rapidly, to a maximum at what is called the critical speed; then the other tendency predominates, and if the shaft is flexible or the bearings a little loose, the body will spin quietly on its true axis of symmetry.

At the speeds usual in large multiple-stage turbines, the degree of precision reached with good machine-shop work gives a sufficiently accurate balancing for practical freedom from vibration.

(*f*) BEARINGS FOR TURBINES. — In the characteristics of high speed but of a generally uniform load to be supported, the turbine is under conditions which belong rather to the electric generator than to the steam engine, and the design of its bearings follows the lines of the former machine. At very high velocity, true alignment and geometrical perfection of the rubbing surfaces are of the first importance, as insuring a uniform distribution of pressure; and with these there must be ample lubrication, under some pressure and supplied by an oil-pump system. In turbines of any size the bearing shell proper is solidly held in the framework of the machine, as indicated in Fig. 357; but there is

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frequently a spherical seating to permit self-alignment with the axis direction. The bearing on the inner side of the De Laval wheel — see Figs. 18 and 363 I — is free to move sidewise also; but a closer examination shows that this is not a bearing at all, in the sense of supporting the shaft, but serves rather as a stuffing box.

The design illustrated in Fig. 361 has several special features. It represents the smaller and quicker-running Parsons turbines, where some flexibility is secured by placing three thin cylinders between the inner and outer shells of the bearing; these are not tightly fitted, and the oil films which form between them have a cushioning effect. The collar bearing holds the rotor against endwise motion, with the particular purpose of preventing side contact and rubbing in the groove

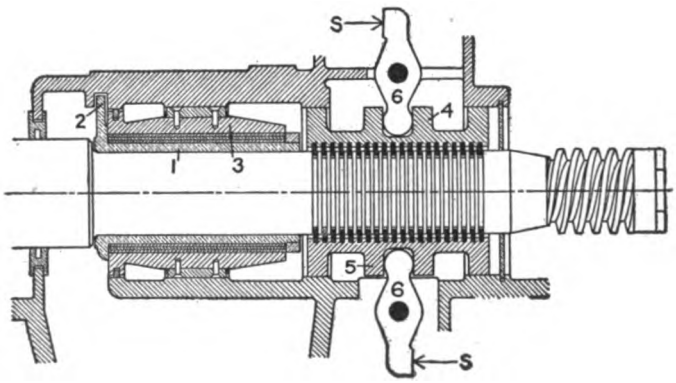

FIG. 361. — Bearing at Governor End of Westinghouse-Parsons Turbine — at right-hand end of Fig. 25.

or labyrinth packing of the balance pistons. The upper and lower halves 4 and 5 are separate, and are adjusted in opposite directions by micrometer set screws on the lines S, S. The worm at the right drives a cross spindle, which carries a bevel gear for the vertical governor, a small eccentric for the admission-valve mechanism (shown in Fig. 378), and a crank to drive the oil pump.

The pivot bearing used at the bottom of large vertical Curtis turbines is shown in Fig. 362; it differs from the simpler arrangement in Fig. 23 chiefly in that the bearing is wholly cut off from possible communication with the steam space. The face blocks are of hard cast iron, disc D keyed to the shaft, C held in the frame, with vertical adjustment by set screws. Oil at high pressure (perhaps 600 to 800 lb. per sq. in.) is pumped in at G, filling the recess between the blocks and forming a film which keeps the annular contact surfaces from really

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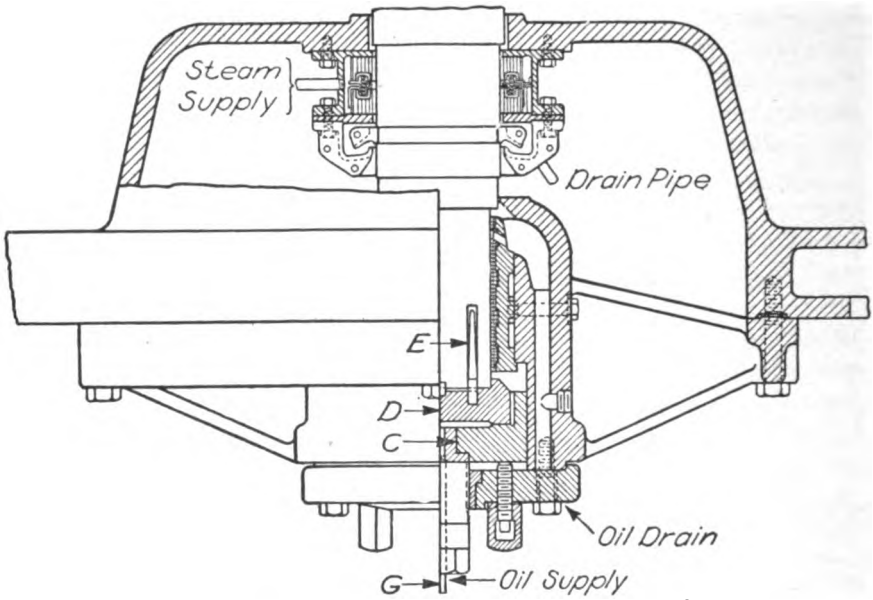

FIG. 362. — Pivot Bearing of Curtis Turbine, from catalogue.

touching. The whole weight of the rotor is thus floated on oil, and the frictional resistance is very low. The arrangement in Fig. 23 is intended for water instead of oil in the bearing, since it would be almost impossible to keep oil out of the exhaust space.

(*g*) STUFFING BOXES. — This name, derived from the engine and implying close contact and pressure of the packing upon the rod or spindle, is applicable to the corresponding part of the turbine as describing the function of preventing leakage rather than the manner of performing that function. Actual contact is used in a few cases; but the more generally accepted scheme is based on the idea of avoiding the contact of rapidly moving surfaces, while yet allowing but a very narrow (and frequently a very crooked) passage for the flow of steam. A number of special devices and methods are used; as will be made clear by the examples collected in Fig. 363.

Plain contact rings are seen at the top of Fig. 23; these rings are made of carbon, are held in place by brass skeleton frames, are pressed inward by light helical tension springs, and bear upon a brass bushing on the spindle. Referring to Fig. 347, which is fairly typical of Curtis-turbine conditions, we see that this packing will have to retain a pressure of no more than 50 or 60 lb. above atmosphere; and with the multiple effect of a series of rings, leakage can be prevented with only a comparatively small contact pressure. Another example of simple contact

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is given in Fig. 363 I, where air must be kept out when the turbine is running on vacuum. A passage can be opened only by the forcing out of the film of oil between journal and bearing, which would require more than a pressure of 12 or 13 lb. per sq. in. There will evidently be, however, some tendency for the oil to work gradually into the steam space.

The essential feature of the scheme shown at II and III in Fig. 363 is the annular space S_1, kept filled with steam of nearly atmospheric pressure by means of a reducing valve from the steam supply and a relief valve to the condenser. There will be a gradual flow through the very narrow passage permitted by the sleeve at B, outward to S_1 at the high-pressure end, inward from S_1 at the low-pressure end of the turbine — all the "stuffing boxes" being piped to form a common system. In effect, the turbine is surrounded by an atmosphere of steam at the points where leakage can occur; and then this steam atmosphere is shut off from the air by lightly fitting segmental contact rings, tied together by helical band springs and pressed inward axially by compression springs which exert the force marked *F*. Oil is supplied at L. The design at III adds labyrinth packing, both in the stuffing boxes and between the wheel chambers — compare Fig. 20.

A different idea is shown in Fig. 363 IV, which is used to prevent

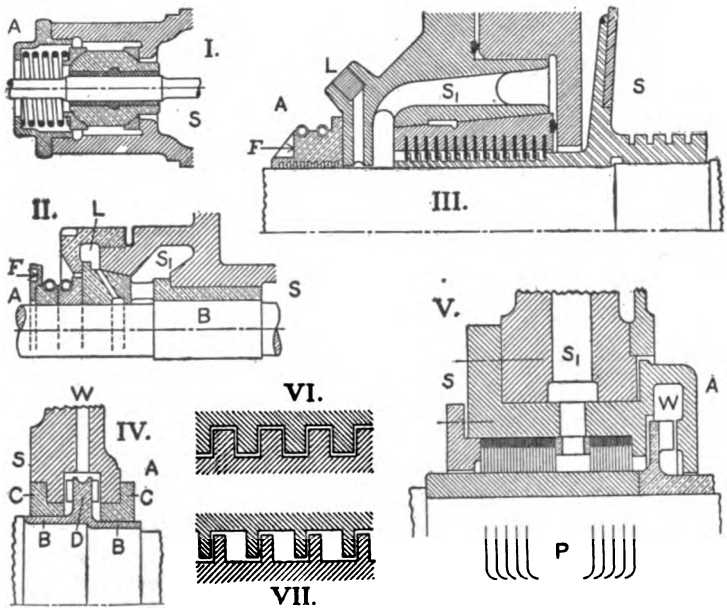

FIG. 363. — Packing Devices for Turbines: I, De Laval, from Fig. 18; II, Rateau, as in Fig. 20; III, Oerlikon-Rateau, later design; IV, Parsons, from Fig. 25; V, Sulzer, Fig. 26; VI, VII, Two Types of Labyrinth Packing.

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the leakage of air into a steam vacuum. There is, of course, a fairly close running fit between the bushing B and the rings C, C; but the passage is closed or sealed by water in the annular space about the collar D: this water has a dynamic pressure, acting radially outward, and produced by little vanes on the sides of D, which are formed like those on the "impeller" of a centrifugal pump. This arrangement has the further advantage that by circulating the water the stuffing box is kept cool.

In V the last two ideas are combined, together with a special form of labyrinth packing, made of thin brass plates. These plates or washers, only about 0.005 in. thick and separated by narrow rings of thicker plate, are bent or flanged where they touch the shaft, as shown by the enlarged detail below the main sectional view.

Labyrinth Packing. — This device, used mostly for preventing internal leakage, is made in a wide variety of detailed form, but the idea is sufficiently represented by the plain outlines at VI and VII in Fig. 363. The first scheme merely provides a narrow, tortuous passage for the steam, of nearly uniform cross area; the second and far more effective type adds the further retarding influence of large and abrupt changes in area of channel. In a long, narrow, straight channel, surface friction alone would hinder flow; and the introduction of sharp bends (type at VI) gives a much higher resistance. A multistaging of the pressure drop (type at VII), with alternate generation of velocity and dissipation in eddies, greatly retards the escape of steam through a channel of a certain minimum cross area.

(*h*) NOZZLES AND DISTRIBUTORS. — As regards the construction of these parts, one typical method is represented by the De Laval nozzle, which is made as a separate piece and has the form given it by the operations of turning, drilling, and reaming. It is held in place by friction (plus steam pressure) and can be pulled out by means of a withdrawing tool that screws over the inner end. The shut-off valve is a part of the scheme of power control: for light loads, the number of nozzles in service is decreased, instead of letting the governor throttle the steam excessively. In larger turbines, groups of two to four nozzles are beneath one hand valve. Another example of nozzles made separately is seen in Fig. 354; but after machining these were bent to curve, and at the mouth were pressed to a square cross section.

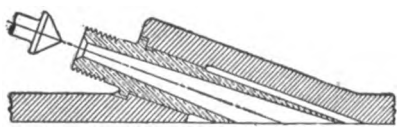

FIG. 364. — De Laval Nozzle, with Shut-off Valve.

High-pressure nozzles for Curtis turbines, with a slight divergence as in Fig. 21 or Fig. 320, are cored out in the cast nozzle segments —

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see first two stages in Fig. 23, and compare diagram of admission ranges in Fig. 22. These nozzles are round in entrance and throat, but change to a squared section toward the mouth; they are smoothed by filing and are planed out in the rectangular part by a special machine. The low-pressure distributors, profiled as in Fig. 19, are made with pieces of ordinary sheet steel, which are set in the cores and cast right into the diaphragms: heavy cross struts (like very short wheel arms), above these nozzle plates, connect the body of the diaphragm to its rim.

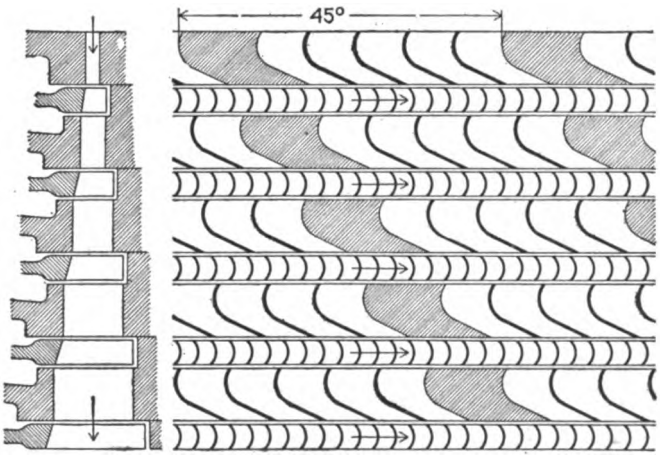

FIG. 365. — Developed Sectional View of Multistage Impulse Turbine, with full peripheral admission: outline of the low-pressure part of a Zoelly turbine.

Figure 365 is intended to illustrate the last point, showing the heavy cross struts which, at intervals around the circle and for strength, must

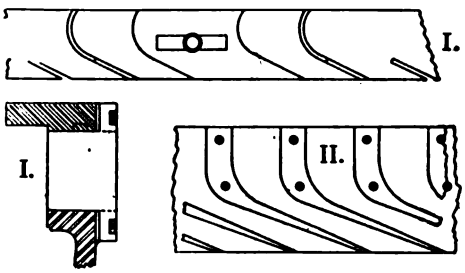

FIG. 366. — Built-up Nozzles. I, Zoelly design, Fig. 365; II, Type of Sulzer, Elektra, etc.

join the rim and body of the partition disc. The detail of construction is shown in Fig. 366 I. Inclined slots are milled (sawed) in the rim of the diaphragm wheel and the edge of the cover ring, and the straight part of the division plate has projections which go into these slots and

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are held by light retaining rings. On the cross struts are short tenons to resist the steam pressure difference on the diaphragm. The analogous construction of nozzles for large pressure drop is indicated at II; the partition pieces are fastened between cylindrical surfaces in an axial-flow turbine, between plane surfaces when the admission is radial.

(*i*) VANES OR BLADES. — There is a wide variety in the form of these essential working parts, and a few typical examples will now be given. The De Laval buckets, Fig. 367, are drop-forged from fairly hard steel and machined on the contact surfaces, their small number permitting this rather expensive method. The steam surfaces are left with the forge finish, except that the entrance edges are ground sharp. A dovetail fastening is used to resist the very high centrifugal force on the blade, and the form is such that any single blade can be removed without disturbing the others. To minimize the possible damage from a bursting wheel, the disc is cut thin just inside the rim, with the idea that the latter will fly off in small pieces before the whole wheel will burst from overspeeding.

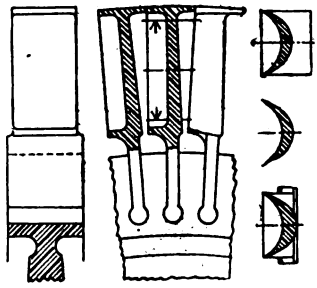

FIG. 367. — De Laval Turbine Blade.

Two other designs using the dovetail fastening, but intended for force conditions (due to speed) less severe than those in the De Laval

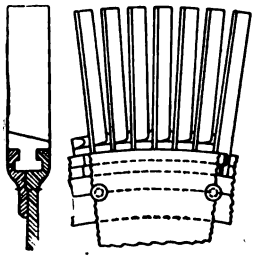

FIG. 368. — Zoelly Blading.

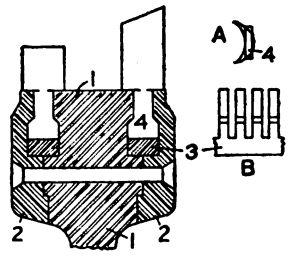

FIG. 369. — Sulzer Blading.

turbine, are shown in Figs. 368 and 369. Here the blades are made from drawn and polished bars of high-nickel steel, and the most important difference between the two schemes is seen in the method of spacing the blades. Distance blocks, cut from a properly-shaped bar and machined with the blades so as to fit the dovetail slot, are used in the first case. In the second there is a spacing ring, shown especially at B: the roots of the blades are hot-pressed and flattened, as best made evident at A, and will then slip into sawed slots in the ring 3.

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The quite common use of vanes formed from rolled plate, by bending or pressing, is illustrated in Fig. 370, together with a method of attachment to the wheel by rivets and with a band or shroud ring to cover the outer ends of the vanes, spacing and steadying them. The form of connection shown in Fig. 371 is far stronger and more durable:

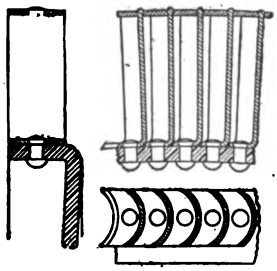

FIG. 370. — Older Rateau Blading, with flanged disc.

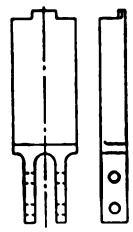

FIG. 371. — Present Rateau Blading, forked over edge of disc.

the blade may be either drop-forged as here sketched, or (more usually) pressed from sheet metal.

In some older Curtis turbines, the blades were cut from the solid metal of the wheel, as shown at I in Fig. 372. The cutting tool travels on a circular path, the tool bar either turning continually or having a back-and-forth rotation; in either case, this bar must be drawn back-

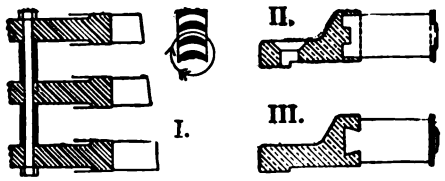

FIG. 372. — Curtis Blading.

ward in the direction of its axis during the idle part of the movement, so that the tool will clear, then advanced for the cutting stroke. This method, however, is not very satisfactory, and the blades are now made from metal bar of proper shape: two schemes for holding them are sketched at II and III. In the first case, the blades are cast into the segment of base ring, being held in a core-sand facing which forms one side of the mold; they are made of a brass composition soft enough to fuse into solid union with the ring. Sketch III shows a simple dovetail fastening, with distance blocks as in Fig. 368, very easy to apply when the holding rings are made in a number of short segments, as is the usual practice.

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(*j*) BLADING FOR REACTION TURBINES. — The method which has been most used for holding the blades of the Parsons turbine is sketched in Fig. 373. The blades are cut to length and simply set into the grooves, with distance blocks of proper profile between them. These blocks have parallel sides (vertically), but are made of a soft metal; and after a whole ring has been filled, the blocks are calked, by means of a tool which reaches down between the blades, and are thus pressed out, more or less perfectly, into the dovetail slot. To give a better grip on the blade, a couple of notches are made in the back of it, into which the metal of the block will be forced. The blades in the casing are held in the same way, except that plain slots are used, not dovetailed. The ends of the blades are left free, but the very long blades of the low-pressure stages must be braced against a tendency to get into vibration. One scheme is to cut a slot in the entrance sides and solder in a wire ring, as at A in Fig. 373: to obviate undesirable effects of expansion this ring will be made in several separate segments, instead of being continuous all the way around.

The scheme shown in Fig. 374, used in this country by the Allis-Chalmers Company, has a spacing and holding ring, with slots of

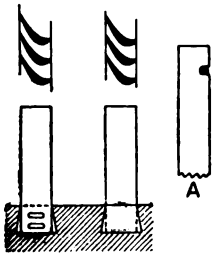

FIG. 373. — The Standard Parsons Blading.

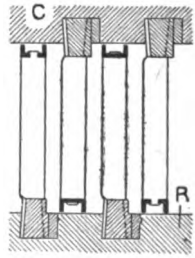

FIG. 374. — Williams-Robinson Blading for Parsons Turbines.

special form into which the ends of the blades fit: these ends are pressed into dovetail form, so that the blades are securely held. A narrow locking ring presses the main ring into the dovetail; and on the rotor R this extra ring is calked into a little side groove. The shroud ring, of channel-bar section, holds the blade ends securely, and can be accurately turned off, so as to make the running fit uniform and as close as is permissible, this minimizing leakage.

(*k*) WEAR ON VANES. — The blading is the weak point in a turbine, because it is the part most liable to rapid wear and deterioration. A steam jet of very high velocity exerts considerable erosive action, greater if the steam is wet than if superheated, and especially aggravated when particles of solid matter are present, as when boilers prime

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on dirty water. Breakage of blades is a not infrequent accident in reaction turbines, especially at the low-pressure end, where the blades are very long. This may result from vibration under steam action, or from rubbing at the ends due to distortion of rotor or casing; while if water has a chance to collect and be thrown among the vane rows in a considerable body, it has enough mass to act as a destructive "foreign body," at the speeds existing.

The materials used for blades or vanes range from various brass or bronze compositions, through common low-carbon steel, to a nickel steel with so high a proportion of nickel that it will be practically rustless. The Parsons type with its tremendous number of blades calls for material that can be easily worked, and with its low steam velocity permits a comparatively soft material. The blade bars, of a suitable alloy (yellow metal), are generally made by the "extruding" process, semifluid metal being forced out through a die and cooled to a solid as it issues — after the manner long used in the manufacture of lead pipe. The turbines with fewer blades and higher velocities require harder materials, but can stand a higher cost per unit part.

(*l*) GOVERNING THE TURBINE. — Three systems of power control may be distinguished: the first is plain throttling; the second, admission

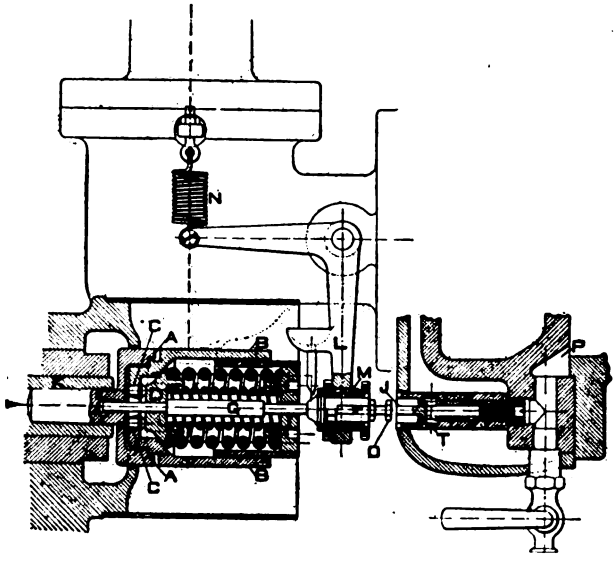

FIG. 375. — De Laval Governor and Vacuum Breaker.

in puffs; the third, variation of initial nozzle opening. One example of each type will now be presented.

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The governor and admission valve of the De Laval turbine, Figs. 375 and 376, will serve as the first example: for although this turbine, through use of the hand valves described in Art. (*h*), under Fig. 364, may enjoy in a large degree the third type of control just named, its automatic regulation is entirely by throttling. An external view of the governor appears at H in Fig. 18, which shows how it is carried on the end of the power shaft. The centrifugal weights, hollow semicylinders in form and marked B, B, in Fig. 375, have knife-edge bearings at A, A, in the shell or body E. As the weights swing out, they push the block D to the right against the spring, the fixed abutment I for the latter being screwed into the outer end of the shell. In normal running the spindle G moves the lever L, against the light spring at N, and thus raises and lowers the double-disc throttle valve shown in Fig. 376. Since the valve may not be tight even when down on its seat, additional security against a runaway is provided for in the vacuum breaker. At an excessive speed the spindle G can move H in the lever L, compressing the spring M until D pushes J inward; this opens the valve at T, admitting air to the exhaust chamber, and thus checking the wheel by friction.

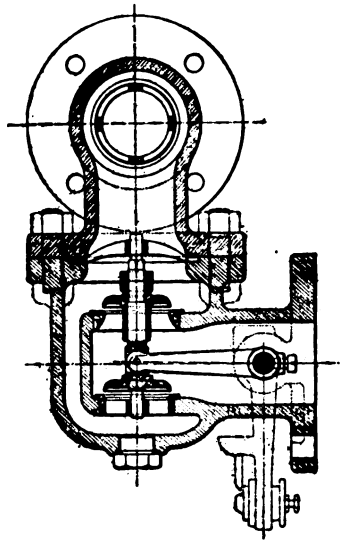

FIG. 376. — De Laval Governor Valve.

(*m*) PUFF GOVERNING. — The original type of Parsons valve gear is shown in Figs. 377 and 378, which belong to the Westinghouse-Parsons turbine in Fig. 25. The valve V_1 in Fig. 377 is not held fast by the governor at the particular height which will make the opening just large enough, but is given a continuous oscillating movement, so as to admit the steam in puffs. The governor mechanism, outlined in Fig. 378, moves the little pilot valve F, which opens and closes the exhaust port E. When E is closed, steam coming through A lifts the piston C, opening the valve; when E is open the steam escapes from the cylinder more rapidly than it can get past the adjusting valve B, and the spring H pushes the valve down. Piston G has a dashpot action, while lever K is for opening the valve by hand, so as to prevent sticking fast when the turbine is standing idle. Under light load the valve V_1 is shut during the greater part of the oscillation period; with heavy load the oscillation may entirely disappear because valve F will not

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open port E at all. The steam passing through this controlling system need not be wasted, but can be returned to the turbine at a lower-pressure point. The bypass valve for overload, V_2 in Fig. 25, is actu-

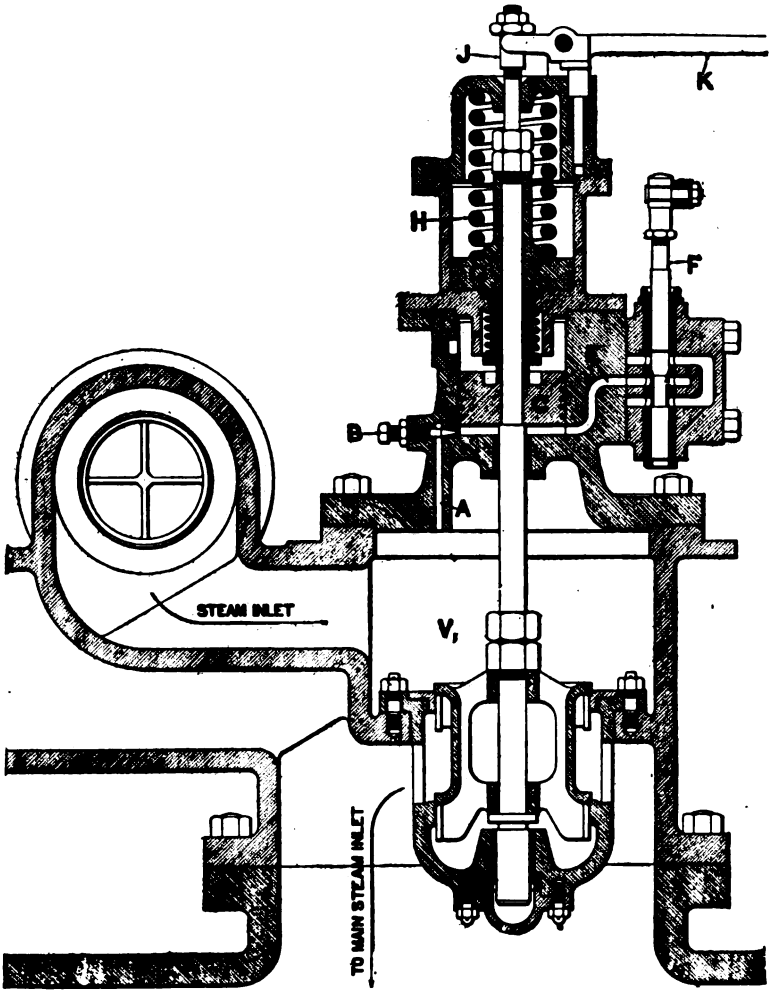

FIG. 377. — Parsons Admission Valve and Controlling Mechanism: this is valve V_1 in Fig. 25.

ated by steam in the same manner, but its pilot valve is given only a simple displacement, without oscillation.

The governor mechanism is sketched in Fig. 378. The eccentric E is on the same spindle with the wheel driven by the worm shown in Fig. 361, and the lever 1 is oscillated once to every so many revolutions,

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say 5 or 6, of the rotor. The governor being at one end of the machine, piece 3 takes the form of a long rod or shaft, with arms keyed to it. The function of the governor is simply to raise and lower the fulcrum A.

In this system of control, the turbine has a periodic pulsation of admission pressure. Steam-channel areas and vane angles have been so designed, presumably, as to give best performance with steam of full pressure. Most of the steam used enters during the high-pressure portion of the cycle period, hence works to better effect than if at the

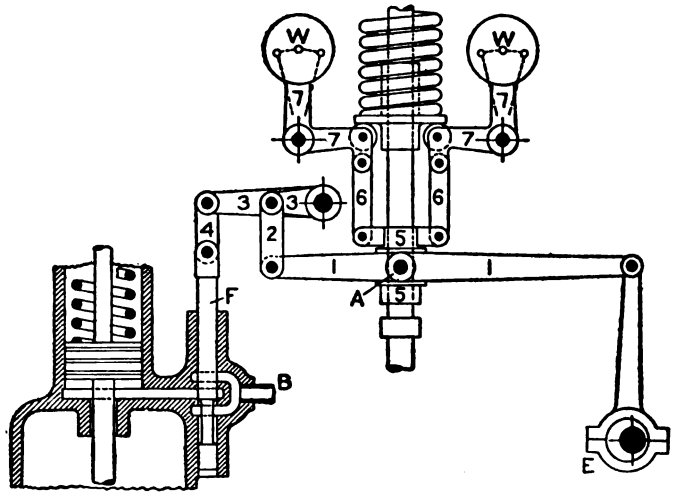

FIG. 378. — Outline of Governor and Valve Gear, Westinghouse-Parsons Turbine.

equivalent mean steady pressure. Against this argument must be set the fact that a part of the steam works at very low efficiency; while the alternate acceleration and retardation of the whole steam current certainly seems likely to exert a harmful influence. No data are available as to the manner in which the pulsations extend through the turbine to the low stages, but there will evidently be a tendency to damp them out. In a many-stage impulse turbine, with its large enclosed steam spaces in the wheel chambers, such modulating action would be very strong — but the scheme is never used with that type of machine.

As between the two systems of puff admission and steady throttling, no comparative data are extant: the difference in net results is probably so small that repeated and very careful tests of the same machine would be needed to establish its existence and amount. One undoubted merit of the Parsons device is, that since the valve is in continual motion it is never likely to stick fast and perhaps cause the governor to lose control. Most builders of Parsons turbines have used

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this valve action, with variations in mechanical detail of the driving apparatus, but some have gone to plain throttling. Concerning mixed-type design, it is enough to cite the facts that the Westinghouse Machine Company have carried puff governing into their double-flow turbine, while the Sulzer turbine (of which Fig. 26 is an early example) is governed by throttling.

(*n*) CUT-OFF CONTROL. — The scheme of automatically regulating the number of first-stage nozzles open for admission was first worked out and applied in the Curtis turbine. The inlet valves, generally about eight in number and each controlling a small group of nozzles, stand in a row in a steam box or chamber, S in Fig. 379; this valve box subtends but a small arc of the turbine circumference, and in the larger machines is one of a pair. At I is shown a cam mechanism for lifting the valves, at II a steam-actuated valve. The governor works, of course, on the relay principle: acting through a self-centering gear essentially equivalent to that shown in Fig. 290, it moves the slide valve of an oil-operated "hydraulic" cylinder. From the crosshead of the latter comes the rod H, which turns the cam shaft B; and the cams along this shaft, one to a valve, are set in series, each a little later than the preceding one, or a certain number of degrees behind it. Under any particular load there will be a number of valves wide open, one partly open, the rest closed; the single valve that is just in the act of opening for any position of the cam shaft gives the close gradation of power.

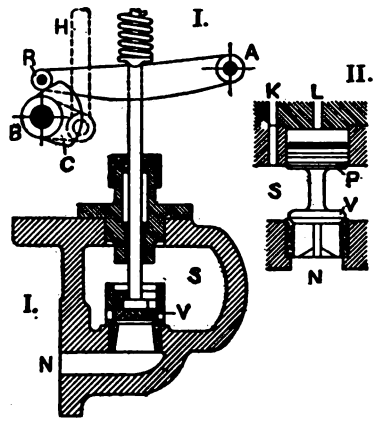

FIG. 379. — Nozzle Valves for the Curtis Turbine.

With steam-actuated valves, as at II, the governor operates a series of small pilot valves, one of which determines whether steam of full pressure shall be admitted above the piston P or whether this space shall be opened to exhaust. In the latter case the valve will be lifted, because the piston is larger in diameter than the valve disc. The pilot valves are generally moved by a cam shaft like that here used for the main valves; while magnet lifts have also been employed.

The governor, shown in Fig. 380, is carried on the top of the main spindle, above the generator. The weights A, arranged very much as in Fig. 375 and similarly pivoted on knife edges, pull at B on the tension spring D. Rod C, which turns, of course, with the rest of the

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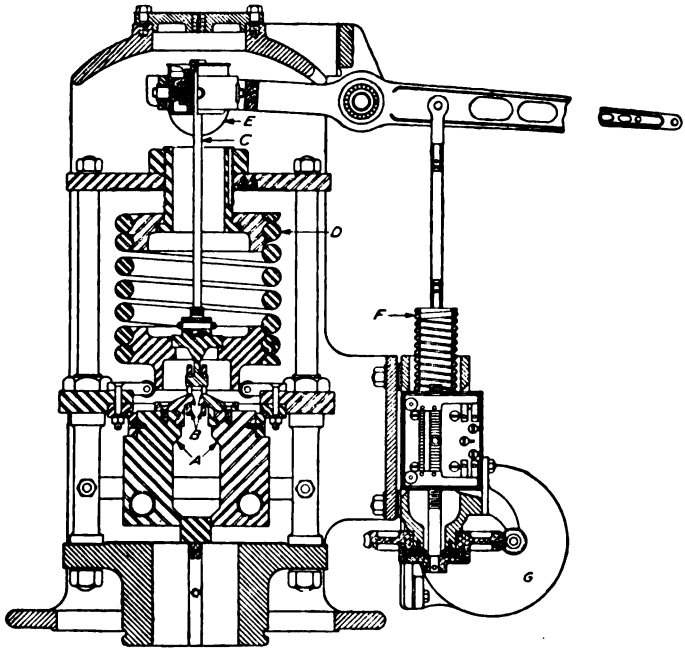

FIG. 380. — Governor of the Curtis Turbine.

governor, is connected by a ball-bearing joint at E to the stationary lever, the outer end of which moves the pilot valve of the oil cylinder. At F is an auxiliary spring for changing the running speed, which may be adjusted from the switchboard by a little motor G.

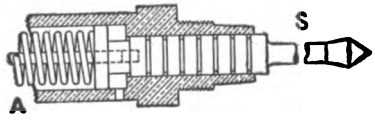

FIG. 381. — Self-acting Nozzle Valve for De Laval Turbine.

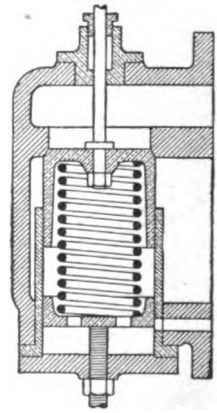

FIG. 382. — Automatic Bypass Valve for Curtis Turbine; enlarged from Fig. 23.

Self-acting Valves. — In Figs. 381 and 382 are given two examples of valves operated by pressure differences which arise within the turbine. The first is an automatic substitute for the hand valve shown in Fig. 364. With full pressure in the valve chamber, the grooved plunger is pushed to the left against the spring; but when the governor

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throttles the steam, the spring force predominates and closes the valve. For the position and connections of the inter-stage valve in Fig. 382, refer to Fig. 23. It serves the special group of second-stage nozzles marked N_B in Fig. 22, opening when, because of heavy load and a large initial admission, more steam gets in than can be accommodated by the regular second-stage nozzles. With pressure from the first wheel chamber on top and that from the second chamber below, the valve opens when the difference between these exceeds the push of the spring.

(*o*) Effect of Governor Action. — The question, What is the effect of a change from normal load, and how does it differ in the two cases of throttling and of cut-off control? will now be briefly considered. As characteristic of normal-load running, we assume that in all the stages the steam velocity bears the same ratio to the vane speed, according to § 46 (*j*), and that the final drop to exhaust pressure just gives to the last stage its proper share of energy. The chief determinant of this action is the manner of variation of the channel section, as pointed out in § 49, and at each point along the channel there will be a characteristic normal pressure, with which the local pressure under changed conditions may be compared.

The most obvious result of a decrease in the amount of steam admitted is to lower the local pressure all along the line, while an increase will raise it. This will give the last stages a smaller proportion of the total energy development at light load, a larger share at heavy load. Under very light load the exhaust pressure tends to creep up into the turbine, the gradient through the last stages being only enough to maintain the flow; under heavy load, high pressure backs up toward the entrance.

Now plain throttling, which cuts down the steam pressure at entrance, tends to keep the division of work among the stages more nearly equal at light loads — while diminishing, of course, the total energy available. The cut-off method admits steam at full pressure, but it has a big drop through the initial stage, and thereafter the local pressure will be about the same as with the first manner of control. The advantage of the second method depends upon the ability of the first stage to absorb effectively the relatively large energy made available for it; but since such a capability will always exist to some degree, it seems that the scheme of having all the pressure drop within the turbine itself is inherently better than that of allowing a considerable part of the drop to take place in the governor valve.

Control of successive nozzle areas, most practicable in few-stage several-impulse turbines, has not been carried in the Curtis turbine beyond the use of the bypass valve in Fig. 382: it is held that the possi-

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ble small gain will not overbalance the disadvantage of greater mechanical complexity. Of the Schulz turbine represented by Fig. 348, quite extensive tests are described in the reference from No. 34 in Table 20: naturally, better economy resulted from an equable division of work among the stages than from an irregular distribution. Puff admission has somewhat the effect of cut-off control, in that most of the steam enters at higher pressure, as just pointed out in Art. (*m*). With the resulting fluctuation of driving force, the fly-wheel effect of the heavy rotor is required for the maintenance of steady speed.

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CHAPTER XI

SUNDRY STEAM APPLIANCES

§ 52. Steam Jet Apparatus

(*a*) Action of Entrainment. — The steam jet from a nozzle, beside being applied to the driving of turbine vanes, may be made to impel a stream of air or water. The steam discharges into a space filled or supplied with the substance to be moved; there it picks up or entrains a certain amount of this substance, forming a mixed jet of greater weight but much smaller velocity; and then this resultant jet is discharged through a suitable retarding nozzle against a pressure higher than that at the mixing point.

These devices are, mechanically, the simplest used for applying the expansive energy of steam to work performance, since they have no moving machine parts. They are, however, of very low efficiency: large wastes of kinetic energy occur in the mixing operation, and in the retardation of the current against rising pressure there are further losses of effect. Besides, to insure delivery, the discharged jet must have a very considerable excess of velocity and energy, which is necessarily dissipated into heat as it comes to rest.

(*b*) Steam Blowers, used for producing draft for boiler furnaces, are of two types. The first is the exhaust jet of the locomotive, in which the whole body of steam used by the engine mixes with the products of combustion, drawing them into an enlarged jet which is expelled up the smoke stack. These gases, coming from the boiler tubes into the smoke box, are at a temperature much above that of the exhaust steam: consequently the steam is not condensed, but is rather dried and superheated, and we have the case of two gases mixing. A typical example is outlined in Fig. 383, a good deal of nonessential detail being omitted in the drawing, especially the spark-arresting screens; these serve also to break the force of the current from the tubes into the space above the exhaust nozzle N, and to deflect the greater part of it downward, so that the jet draws mostly from below. This is from a large locomotive, where the stack has to be let down

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into the smoke box, and a dead space is left above the horizontal partition P.

The second type of blower is illustrated by the example in Fig. 384, where a small jet of high-pressure steam is used for moving a stream of air relatively much larger than that in the locomotive (as compared with the weight of the steam), but against a much smaller resistance. Further, air of ordinary temperature is drawn into this blower, so that the steam will be partly condensed in the formation of the

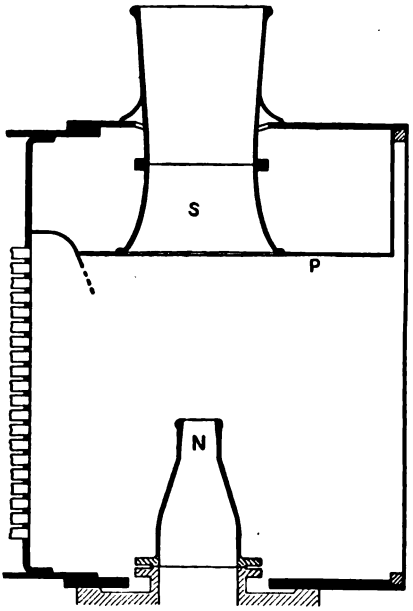

FIG. 383. — Locomotive Smoke Box.

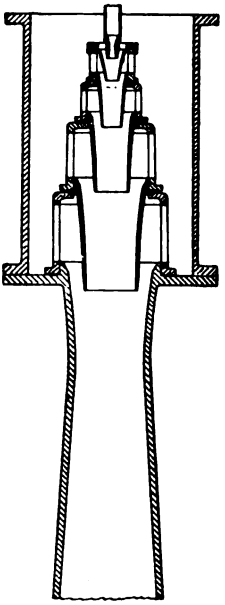

FIG. 384. — Forced-draft Blower.

mixed jet — the action approaching that of the injector in this respect. Note how the entering air is split up into several concentric divisions, with the intention of reducing the eddy loss during the mixing operation.

The following example, worked out with assumed data, will show the method of computing the efficiency of this sort of apparatus in actual operation, and will give some idea of its probable value for the first type:

EXAMPLE 54. — In a locomotive boiler, 1 lb. of coal will evaporate about 6 lb. of water, and the gaseous products from 1 lb. of coal will weigh about 18 lb., so that there will be 3 lb. of gas to 1 lb. of exhaust steam. We assume that

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the mixture of hot gas and superheated steam will have a temperature of 400 deg. fahr., and that the gas alone has the same density as pure air. For air at 400 deg., disregarding the small variation from atmospheric pressure in the smoke box, the specific volume would be, by Eq. (12),

$$v = 12.4 \times \frac{860}{492} = 21.7 \text{ cu. ft.}$$

At the same pressure and temperature, the pound of steam measures 34.7 cu. ft., by Table VI: then the whole volume discharged through the stack, per pound of steam, is $34.7 + (3 \times 21.7) = 99.8$, or say 100 cu. ft.

The static resistance overcome is the pressure difference, or the "vacuum" in the smoke box, which may be assumed as equivalent to 4 in. of water column. One foot of water column equals 62.4 lb. per sq. ft., so that the resistance is 20.8 lb. per sq. ft.; and the work done in expulsion will be $100 \times 20.8 = 2080$ ft. lb.

Now the steam is expelled from the cylinder and forced through the nozzle by the back pressure upon the piston, over and above the atmosphere, together with the available work of the uncompleted expansion, represented by the triangle CHD in Fig. 57; this may easily amount to 4 lb. per sq. in. through the whole steam volume; and taking the latter to be 24 cu. ft., with allowance for the condensation due to work done in the cylinder, we get about

$$4 \times 144 \times 24 = 13{,}820 \text{ ft. lb.}$$

as the energy of the steam jet per pound of steam. Then the mechanical efficiency of the apparatus is

$$E = \frac{2082}{13{,}820} = 0.151.$$

Referring forward to Eq. (244), which applies equally well to this case, we find the limit of efficiency in the entraining operation, with our ratio of masses, to be about 25 per cent, the other 75 per cent of the jet energy being necessarily changed into heat: and comparison of the realized 0.15 with the maximum attainable 0.25 gives an efficiency of $0.15 \div 0.25 = 0.60$ for the operation of jet retardation and pressure-work performance.

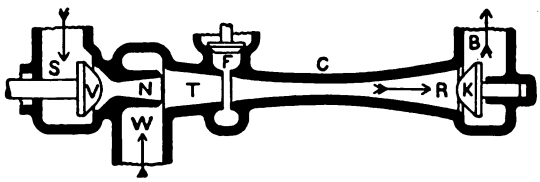

FIG. 385. — The Simple Injector.

(c) THE INJECTOR. — The simplest type of this apparatus is outlined in Fig. 385, where all structural detail is left out, and only essential form is shown. Steam from the boiler enters at S, and its admission to

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the nozzle is controlled by the hand-regulated valve V. To start the injector, this valve is opened a little way, and the steam jet at first draws out air from the water chamber W and from the water pipe, until the vacuum is sufficient to lift the water; as soon as the flow of water is established, steam is turned on full, and the mixed jet, formed by condensation of the steam in the tube T, all reduced to water by the time it gets to the throat C, and then slowed up in the retarder R, develops enough pressure to open the check valve K and force itself into the boiler. The overflow F permits the escape of the mixed steam and air at the start, and of the first water, which is propelled by a jet of steam too weak for the regular discharge. This overflow is located at a part of the tube where the pressure in the jet is less than that of the atmosphere, in normal running; and the automatic check valve at F prevents air from getting in to spoil the vacuum.

The rate of delivery of an injector can be varied over a considerable range — in some cases from full capacity to as little as 40 per cent of that rate. In the simple form, this regulation is made chiefly by choking down the current of water in the suction pipe; something can be done by diminishing the supply of steam, but if this is cut down very much, the jet may become too weak for expulsion, and the injector will "kick back."

(*d*) THE COMPOUND INJECTOR. — The injector with only one tube is not very sure in its action if the supply water must be lifted through

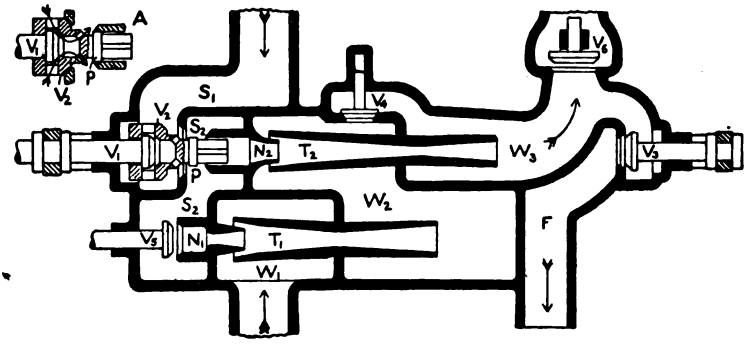

FIG. 386. — The Double-tube Injector.

any considerable height: to overcome this difficulty, two working jets are used, one for suction, the other for forcing. The general arrangement and the form and dimensions of the working parts of the injector outlined in Fig. 386 are copied from an actual design, the "Metropolitan": but besides the omission of details of construction and the

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mere indication of such parts as stuffing boxes, some of the essential parts are transposed from their regular positions, so as to make a clearer illustrative drawing.

The stems of the two valves V_1 and V_3 are rigidly connected by external side bars, so that they must move together, and are controlled by a suitable lever handle at the left. When the injector is idle these are pushed over to the right so that both steam valves, V_1 and V_2, will be closed and the overflow V_3 wide open. To start, the handle is drawn back just a little, bringing the lifting valve V_1 into the position shown at A, and admitting a small amount of steam through the chamber S_2 to the lifting nozzle N_1, so as to establish the suction. The piston P keeps the nozzle N_2 shut off, not only during this preliminary admission of steam, but also until V_2 has been given quite a movement. In fact, V_1 is not essential to the working of the apparatus, but is rather designed to equalize the pressure on the two sides of V_2 before this larger valve is moved, so that there will be no great resistance offered to the moving of the handle. In order to insure prompt filling of the whole injector with water at the start, the overflow is put on the discharge chamber W_3; and the passage through the forcing tube T_2 is supplemented by the check valve V_4.

As soon as a good stream of water appears at the overflow, the handle is pulled all the way back, giving full admission past the main valve V_2 to the steam chamber, and closing the overflow: in the drawing, the injector is not quite wide open, and there would be some waste of water through the overflow valve. The small valve V_5, at the lifting nozzle, is not intended ever to be closed, but is adjusted by hand so as to regulate the rate of delivery by varying the amount of water supplied to the forcing tube T_2.

(*e*) Theory of the Injector. — In all these devices for impelling fluids by means of a jet of steam, there are two principal ways in which the available kinetic energy of the jet is wasted or dissipated: namely, in the operation of mixing or entrainment and in that of retardation. The first loss is necessarily very large, and its amount may be approximately reasoned out as follows:

If a small body moving at high speed impinges upon a larger body moving slowly, but in the same direction, the sum of the momenta, or of the several mass × velocity products, will be the same before, during, and after the impact. If the bodies are nonelastic, they will move together after impact, and there will have been a loss of kinetic energy: for if

$$M_1V_1 + M_2V_2 = (M_1 + M_2)\,V, \quad . \quad . \quad . \quad . \quad (242)$$

then will

$$M_1V_1^2 + M_2V_2^2 > (M_1 + M_2)\,V^2,$$

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or the sum of the original kinetic energies be greater than the energy of the combined mass.* Let E be the latter quantity, and change it from $(M_1 + M_2)\, V^2$ to

$$E = \frac{(M_1V_1 + M_2V_2)^2}{M_1 + M_2}$$

by Eq. (242). Now subtract this from the original energy

$$E_1 = M_1V_1^2 + M_2V_2^2:$$

after reduction, the difference, or the decrease of kinetic energy, is

$$E_1 - E = \frac{M_1M_2}{M_1 + M_2}(V_1 - V_2)^2. \quad . \quad . \quad . \quad . \quad (243)$$

With elastic solid bodies, there is rebound after impact, in which most of the energy that has been absorbed in compressing the solids is restored; with fluids, mingling together, any such action is impossible, and Eq. (243) therefore applies to the operation under consideration. Let M_1 stand for the steam and M_2 for the substance to be moved, which might be called the load; usually the initial velocity and energy of the latter are relatively small, and by dropping V_2 from Eq. (243) we get for energy lost the simple approximate expression,

$$E_1 - E = \frac{M_2}{M_1 + M_2} M_1V_1^2 = \frac{M_2}{M_1 + M_2} E_1. \quad . \quad . \quad (244)$$

In the injector, a good working proportion is 10 or 12 lb. of water to one pound of steam; then the wasted energy will be $\frac{9}{11}$ to $\frac{11}{13}$ and the effective energy only $\frac{1}{11}$ to $\frac{1}{13}$ of that in the steam jet. The latter may have a velocity of 2400 to 3000 ft. per sec. and a kinetic energy of 90,000 to 140,000 ft. lb. per pound of steam (see Table 6), varying with both the initial pressure and that at the mouth of the steam nozzle. Disregarding V_2 in Eq. (242) and using $\frac{1}{12}$ as an average value for the efficiency in mixing, or for $M_1/(M_1 + M_2)$, we have $V = V_1 \div 12 = 200$ to 250 ft. per sec.; and with this V the 12 lb. of mixed jet, or $(M_1 + M_2)$, will have the energy $E = E_1 \div 12 = 7500$ to 11,700 ft. lb. All the rest of E_1 is changed back to heat during the mixing of the fluids.

It is not possible to calculate the efficiency of, or the energy waste in, the operation or retardation and the performance of pressure work; but it is certain that the energy of the jet and its velocity (E and V as just found) must be considerably greater than the work of expulsion

* Omission of the factor $\frac{1}{2}$ from the expression for kinetic energy, a mere matter of present convenience, does not effect the relation deduced and applied.

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against, and the velocity of efflux due to, the final pressure. The last is found most conveniently by the hydraulic equation

$$V = \sqrt{2gh}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (245)$$

To get the effective head h, divide the net discharge pressure by $(61.5 \div 144) = 0.427$: here 61.5 is used instead of 62.4 as the weight of one cubic foot of water in order to take account of the high temperature of discharge from the injector.

Suppose, for example, that the injector feeds a boiler against a gage pressure of 150 lb., and that the water is lifted 10 ft.: then $150 \div 0.427 = 351$ ft.; adding the 10 ft. of suction head and substituting in Eq. (245) we get,

$$V' = \sqrt{64.3 \times 361} = 152.5 \text{ ft. per sec.}$$

If the mixed-jet velocity were $V = 225$, as above, the efficiency in retardation and expulsion would be

$$\left(\frac{V'}{V}\right)^2 = \left(\frac{152.5}{225}\right)^2 = 0.46.$$

(*f*) Test of Injector Performance. — In making a test of the working of an injector, it is enough to measure the water drawn in and to observe the temperature and pressure of suction and of discharge, beside getting the pressure and quality of the steam supplied. The amount of steam used can be found by means of a thermal equation, in which the small quantity of heat converted into work may be disregarded. In the operation of mixing, the heat given off by 1 lb. of steam, above the discharge temperature t_2, is taken up by w lb. of water in being raised from the suction temperature t_0 to this same t_2. Having gotten from the steam table the total heat h_1 of the entering steam, the equation is

$$h_1 - (t_2 - 32) = w\,(t_2 - t_0). \quad . \quad . \quad . \quad . \quad . \quad (246)$$

An example will illustrate the relations involved and the method of getting results.

Example 55. — An injector supplied with steam at 94.5 lb. by gage, with 2.6 per cent of moisture, draws water at the rate of 3240 lb. per hour, showing the temperatures $t_0 = 72.6$ deg., $t_2 = 156.3$ deg.: the supply of water is at 16 ft. below the injector, and the discharge pressure is 99.2 lb. by gage. How much steam is used per hour, how much work of pumping is done per pound of steam, and what is the thermodynamic efficiency of the injector as an engine?

For this steam, at 109.2 lb. abs.,

$$\begin{aligned} h_1 &= 1188.1 - 0.026 \times 883.1 \\ &= 1188.1 - 23.0 = 1165.1 \text{ B.t.u.} \end{aligned}$$

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Then by Eq. (246) the water pumped per pound of steam is

$$w = \frac{1165.1 - 124.3}{156.3 - 72.6} = \frac{1040.8}{83.7} = 12.43 \text{ lb.}$$

Dividing the total water weight of 3240 lb. by this ratio, we find the steam per hour to be $3240 \div 12.43 = 261$ lb.

The pumping work to be credited to the injector, per pound of steam used, is the forcing of 13.4 lb. of total discharge, at 61.0 lb. to the cubic foot against a net pressure of $99.2 + (16 \div 2.31) = 106.1$ lb. per sq. in. Solving by the pressure-volume method of Eq. (23), we have

$$U = 144 \times 106.1 \times \frac{13.4}{61.0} = 15{,}280 \times 0.220$$

$$= 3360 \text{ ft. lb. or } 4.32 \text{ B.t.u.}$$

Now out of a total heat supply of $1165.1 - 124.3 = 1040.8$ B.t.u. (estimated above the temperature of the water discharged), only 4.32 B.t.u. is effectively transformed into work; and the thermodynamic efficiency has the low value

$$E = \frac{4.32}{1041} = 0.00415, \text{ or } 0.42 \text{ per cent.}$$

(*g*) RANGE OF THE INJECTOR. — In the matter of discharge pressure, it is possible, by suitably proportioning the steam nozzle and the inlet to the mixing tube, to have this pressure far above that of the steam, even using the exhaust from a noncondensing engine to feed the boiler. In this case the water pumped per pound of steam will be relatively small, because the energy of the steam jet is small; consequently the supply water must be cold, so that the steam will be condensed without making the final temperature too high. Further, the fact that this temperature must be high precludes the lifting of the water by suction, because the attainable vacuum in the injector will be small.

Ordinary high-pressure injectors are usually proportioned so as to deliver against a pressure from 20 to 40 per cent above that of the steam — that is, in regular working there is a large excess of energy in the water jet over just what is necessary for delivery.

It is frequently desirable to use feed water of fairly high initial temperature, as from the hot well of a condensing plant. The upper limit of suction temperature in a well-proportioned injector will be about 130 deg. fahr. — which in Example 55 would make t_2 about 212 deg. — so that feed at from 90 to 110 deg. is entirely practicable; but of course this water must be supplied to the injector, if of the simple type, at or above its level.

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§ 53. Condensers and Air Pumps

(*a*) PRINCIPLE OF CONDENSATION. — Suppose that a closed vessel has been filled with pure (that is, air-free) water, and that this water has been partly withdrawn; then the space above it will be filled with vapor, of which the " tension " or pressure will be just that corresponding to the temperature existing, according to the relation given in the steam tables. At 120 deg., for instance, the absolute pressure would be 1.69 lb., at 100 deg. 0.95 lb., at 80 deg. 0.51 lb., and so on. The condenser is always a closed space containing water and vapor, but conditions are complicated by the unavoidable presence of a small amount of air; this has entered the boiler in the feed water, has leaked into the engine or turbine or exhaust pipe, or has come in with the cooling water. To maintain a certain vacuum (or low absolute pressure), two things are necessary: there must be an ample supply of cooling water, so that the heat of the steam can be taken up with but a moderate rise of temperature in this water, and the products of condensation must continually be removed. The water of condensation is very easily taken out; but to get rid of the air is a more troublesome problem, as is indicated by the fact that although this air is but a minute fraction (by weight) of the substance handled by the vacuum pump, the latter is commonly called an air pump.

(*b*) TYPES OF CONDENSERS. — The primary division is into the two main classes of jet or mixing and of surface condensers, typically represented by Figs. 387 and 389. In the first, water enters at B and is spread out in a conical spray by the distributor D, which serves also as a regulating valve; steam enters at A, and after condensation in the space just below D the combined stream sweeps down through the neck F into the air pump, and is finally discharged at J. The water current must fill the narrowest part of the channel, below F, and its velocity must be greater than the speed with which bubbles of air can rise through water, so that the air will be effectively carried out, and not allowed to accumulate. The surface condenser, Fig. 389, is a box or tank nearly filled with a closely-spaced body of small brass tubes ($\frac{3}{4}$ in. to 1 in. diameter, outside), through which water flows. To the vacuum pump goes only the condensed steam and the air that came with it, while a separate circulating pump drives the cooling water through the tubes. A jet condenser, if placed not more than 15 to 20 ft. above the supply, will draw its own cooling water by suction; but water under pressure must be available for starting or " priming " the condenser.

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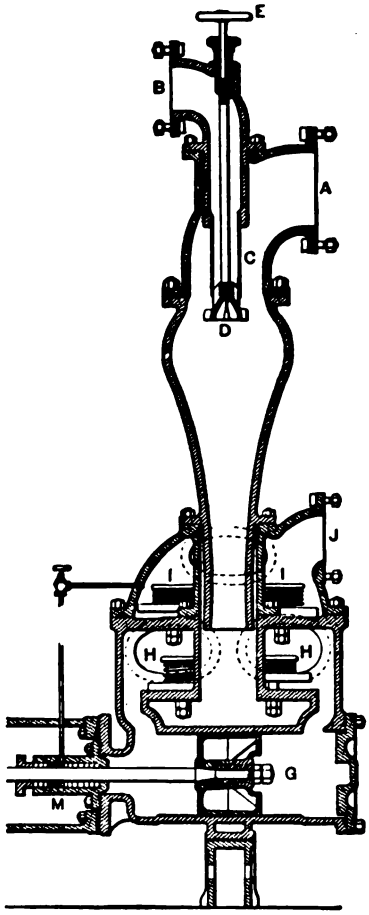

FIG. 387. — Simple Jet Condenser.

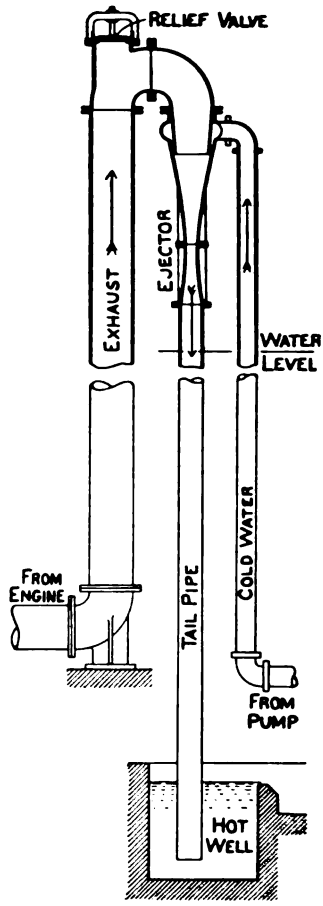

FIG. 388. — Barometric Condenser.

(*c*) REMOVAL OF CONDENSATE. — The jet condenser can be evacuated without the use of a vacuum pump: one scheme is illustrated in Fig. 388, the other, the ejector condenser, can be described in a few words. The barometric condenser is raised so high above the level of the hot well that the top of the column of water in the tail pipe, if determined by the static lift of the vacuum, will be a few feet below the condenser proper — or the latter will be about 35 ft. or more above overflow level in the tank into which the tail pipe discharges. Then the current of warm water flows out by gravity and, with the help of a narrow-necked ejector to insure entrainment, carries the air with it.

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A pump is needed for raising the cooling water, but is helped by the suction lift of the vacuum.

The ejector condenser works on the principle of the injector, the discharging current being given a high enough velocity to carry it out

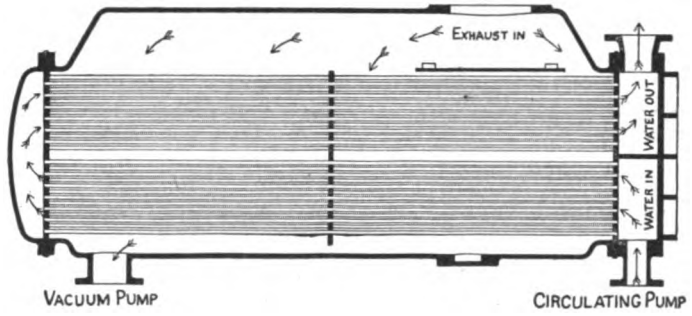

FIG. 389. — Typical Simple Surface Condenser.

against the pressure of the atmosphere. Water enters at the top through a central nozzle, and steam is admitted to and guided against the water stream by a series of narrow conical passages in and around the wall of the mixing tube: in effect, a series of thin annular steam jets impinge upon the stream, and impart some of their velocity to the combined current. Water should flow to this condenser from a supply at or above its own level.

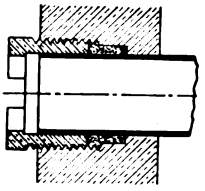

FIG. 390. — Tube Joint for Surface Condenser.

The difficulty of air removal is greatly aggravated by increase of vacuum beyond a certain point, or by lowering condenser pressure to what is desired in steam-turbine practice. Especially, entrainment by the main water current, as in Figs. 387 and 388, becomes less effective and complete as air density decreases and specific volume grows; and similar troubles arise in the method of combined withdrawal from a surface condenser, represented by Fig. 389. To facilitate the handling of large volumes of air and vapor mixture, the two products of condensation are usually separated in a high-vacuum plant. Most simply, water collects alone in a supplementary tank or "hot well" at the bottom of the condenser, as indicated in Fig. 396, whence it is withdrawn by a small pump, of either piston or centrifugal type: this pump may be controlled by a float in the hot well, so as to maintain there a constant water level. A separate "dry-vacuum" pump, built very much on the lines of an ordinary air compressor, now takes care of the air from the condenser. A similar separation of function with a

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jet condenser is shown in Fig. 393: there is some entrainment of air by the current in the tail pipe, but this is supplemented by a dry-air pump.

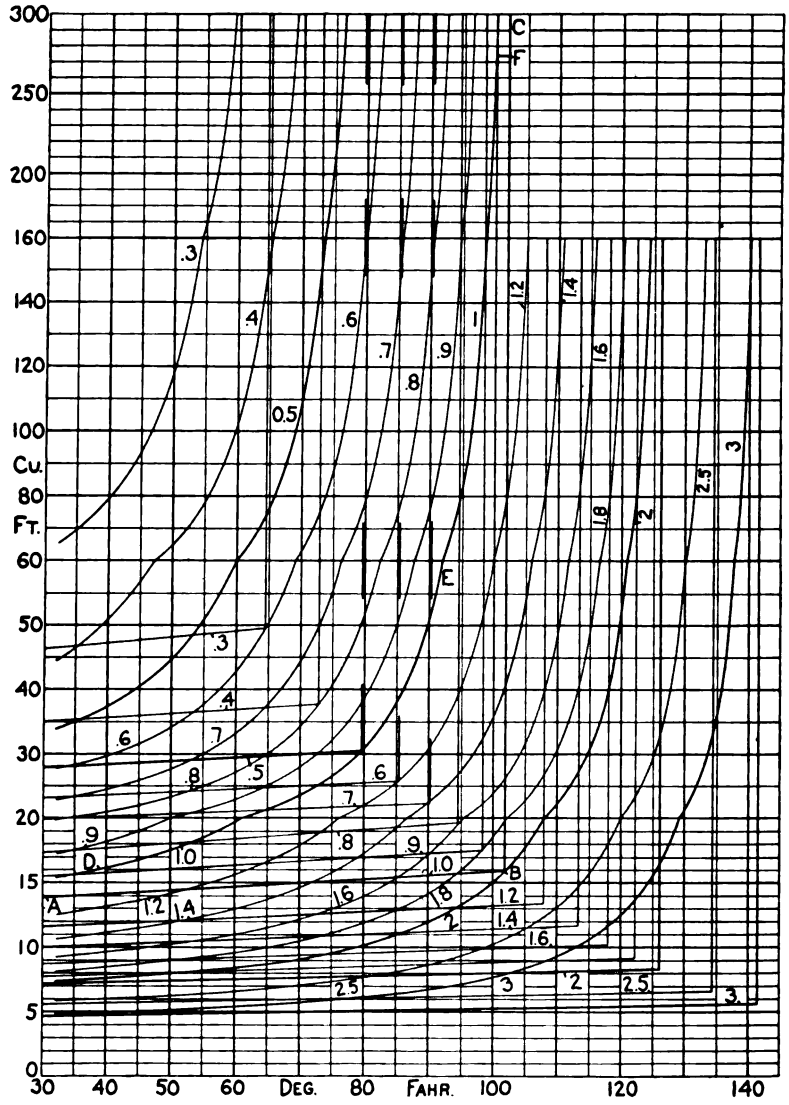

FIG. 391. — Volume of Air and Vapor Mixtures.

The "air" in and from the products of condensation is not really pure, dry air, but is a mixture of the latter with uncondensed water

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vapor. The nearer pressure and temperature come to being in the steam-table relation (or in that of boiling point to pressure), the larger is the ratio of vapor to air in the mixture filling the open spaces of the condenser, not occupied by liquid. As the air mixture is cooled below the steam-saturation temperature for the pressure existing, or as the pressure in the condenser rises above that corresponding with the temperature existing, the volume to be handled by the air pump diminishes rapidly.

(*d*) THE LAW OF GASEOUS MIXTURES. — In general, if air and water are confined in a vessel, equilibrium of composition exists in the space above the water only when the air is "saturated" with vapor. This means that as much vapor will have been formed from the water, and diffused through the air, as is needed to bring the combination to the proportion fixed by the law of gaseous mixtures, with vapor tension as the determinant. In simplest statement, this law is that the product pv for the mixture is the sum of the pv's of the components.* With two gases, it is most obvious to think of them as measured, at pressure p, by the products pv_1 and pv_2, where $v_1 + v_2 = v$. But after diffusion each is spread out through the whole volume v, so that the measures change to p_1v and p_2v, with $p_1 + p_2 = p$. In other words, each part of the mixture exerts a part of the total pressure which is proportional to its relative quantity (by volume, not by weight).

Now let one component be a vapor in presence of its liquid and having a certain specific pressure at the temperature existing: then the mixture (or the gas) will be saturated when the vapor is to the whole as its tension is to the whole pressure. If there be more vapor, as in the exhaust coming to the condenser, it will turn to liquid (without change of pressure or temperature) when the way is opened for escape of latent heat; if there be less vapor, as in the ordinary atmosphere, evaporation tends to take place. Since the mixture in the condenser is always arrived at from the side of excess of vapor, it is sure to be saturated.

Another concept, which underlies hygrometry or the measurement of atmospheric humidity, is that air is saturated when the least cooling (which involves decrease of vapor tension) will initiate the formation of mist, or result in the deposit of dew.

(*e*) VOLUME OF AIR AND VAPOR MIXTURES. — This matter is put into quantitative shape in Fig. 391, which covers the range of condenser conditions. Each curve is drawn for a particular pressure, from 0.3 to 3.0 lb. absolute as marked, and that for 1.0 lb. is lettered, to help the explanation. The basal quantity of air alone is that which

* This is for a physical mixture, distinct from a chemical compound or a solution, as of gas in liquid.

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measures 1 cu. ft. at 14.7 lb. abs. and at 60 deg. fahr. Substituting in Eq. (12), making it $pv = w \times 0.37\,T$, we find the weight of this cubic foot of "free air" to be

$$w = \frac{14.7 \times 1}{0.37 \times 520} = 0.0764 \text{ lb.}$$

Then its volume, with varying temperature at a particular pressure, is

$$v_a = 0.0764 \frac{0.37\,(t + 460)}{p} = \frac{0.02827}{p} t + \frac{13.00}{p}. \quad . \quad (247)$$

At $p = 1$ lb., line AB shows the value of this v_a, and its slight variation with temperature. The right-end limit B is at the temperature of steam saturation for the governing pressure, which is the highest temperature attainable by the air and vapor mixture under that pressure.

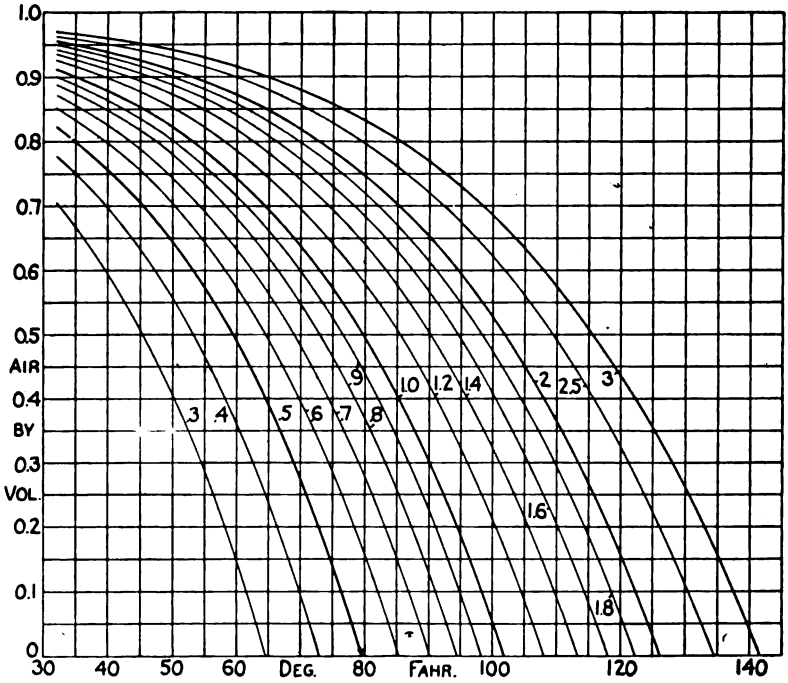

FIG. 392. — Proportions of Air and Vapor Mixtures.

The ordinate distance between line AB and curve DEF represents the volume of vapor that accompanies the air beneath AB, according to the concept that both components are raised to and measured at the full pressure p. At the upper limit BC this distance is infinite, since the pressure-temperature relation of the steam tables is rigorously true only for perfectly pure vapor. Curve FED therefore has the line CB

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as an asymptote; but its very rapid fall to the left of CB shows how greatly the volume of mixture (from the base line to the curve) is diminished by cooling below the temperature of steam saturation.

The ordinate of curve DEF is found by the method of pressure ratios, as set forth in the last article. Let p be the pressure in the condenser and p_s the characteristic steam pressure or vapor tension, at an observed or existing temperature t, lower than that which corresponds with p for steam. Then the whole volume v is to the air volume v_a as the whole pressure p is to the partial air pressure $(p - p_s)$, or,

$$v = \frac{p}{p - p_s} v_a. \quad \ldots \ldots \ldots \quad (248)$$

Another method, based more directly upon the physical state of the air as diffused through the whole space v and exerting the partial pressure $(p - p_s)$, is to substitute the latter pressure in Eq. (12), with the actual temperature, whether for one pound of air or for some other quantity; the resulting air volume will also be the volume of the mixture.

The proportion of the mixture or its ratio of composition is shown in Fig. 392, as a variable with temperature and for the same set of constant pressures that was used in Fig. 391. With the whole height representing unity, the ordinate below a curve, or the fraction of air, is v_a/v or $(p - p_s)/p$; while that from the curve to the top line, showing fraction of steam, is v_s/v or p_s/p.

EXAMPLE 56. — Determine numerical values with $p = 1.2$ lb. for Figs. 391 and 392.

With $p = 1.2$ in Eq. (247), $v_a = 0.02356\, t + 10.83$; and two points, say at 100 deg. $v_a = 2.36 + 10.83 = 13.19$, and at 40 deg. $v_a = 0.94 + 10.83 = 11.77$ cu. ft., are enough to locate the straight air line like AB.

For $p = 1.2$ lb., the boiling point is 108.02 deg. At 100 deg. the vapor tension is 0.946 lb., at 40 deg. it is 0.1217 lb. — these from column 1 of Table II. Then by Eq. (248),

$$v_{100} = 13.19 \times \frac{1.2}{1.2 - 0.946} = 13.19 \times 4.73 = 62.3 \text{ cu. ft.}$$

$$v_{40} = 11.77 \times \frac{1.2}{1.2 - 0.122} = 11.77 \times 1.113 = 13.10 \text{ cu. ft.}$$

The reciprocals of the ratios just used, namely,

$$\text{at 100 deg.,} \frac{p - p_s}{p} = \frac{1.2 - 0.946}{1.2} = \frac{0.254}{1.2} = 0.212,$$

$$\text{at 40 deg.,} \frac{p - p_s}{p} = \frac{1.2 - 0.122}{1.2} = \frac{1.078}{1.2} = 0.898,$$

give air fractions, or ordinates up to the ratio curve in Fig. 392. Remember that these are ratios of volume quantities, reduced to and compared at the same pressure, not of weights.

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(*f*) QUANTITY OF AIR TO BE HANDLED. — This is the most uncertain element in both the design and operation of a condensing plant, since it depends so largely upon rate of leakage. Water coming in from ordinary exposure to the air is likely to contain about 2 per cent of its own volume of free air. By the last term we mean air under atmospheric pressure and at 60 deg. fahr., of which the specific volume is 13.1 cu. ft. to the pound and the density 0.0764 lb. per cu. ft. In what follows all air quantities will be expressed in cubic feet of this free air, which has already been made the basis of Fig. 391.

Stodola (Fourth Edition, page 549) says that we may generally figure on 3 to 5 kg. of air per hour per 1000 kw. of turbine capacity, although citing a determination (of his own) of 35 kg. per hour from a 2000-kw. plant which was in good condition except as to leakage. Allowing 16 lb. of steam per kilowatt-hour, the average rate named (3 to 5 kg.) becomes 0.35 to 0.55 cu. ft. of free air per cubic foot of feed water. Evidently, the 2 per cent, more or less, in the feed water itself is insignificant.

A jet condenser will probably use something like 25 lb. of cooling water per pound of steam condensed. At 2 per cent by volume, this water will carry the free-air equivalent of about 50 per cent of the feed-water volume. The approximate equality here shown between air in cooling water and air in exhaust steam checks up very well with the practical rule that a jet condenser (with both these quantities to be handled) should have about twice as much air-pump capacity as a surface condenser of the same power rating. On this point, Mr. R. M. Neilson * states, for instance, that the air in the injection water may be less than that coming from the turbine, with moderate leakage; but with high vacuum (which generally means a higher ratio of cooling water) the former may be as much as three times the latter. The engine, with stuffing boxes kept well packed, is not likely to have any more air leakage than the turbine.

EXAMPLE 57. — Find the air-pump capacity requisite for a 1000-kw. surface condenser, to maintain a vacuum of 28 in. of mercury or an absolute pressure of 1 lb. per sq. in.

Allowing 16,000 lb. of steam per hour, or 256 cu. ft. of feed water, we take 45 per cent of this, or 115 cu. ft., as the volume of free air per hour. Even in a plain condenser like Fig. 389 the condensate will be cooled quite a little below the temperature of the exhaust steam, since the coldest tubes are at the bottom of the condenser. Let us assume that the final temperature will be 75 deg.; then by Fig. 391 the volume per cubic foot of free air will be about 26.5 cu. ft.

* Factors Affecting Air Pump Capacity, *Power*, Nov. 29, 1910.

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The air pump will therefore have to take in 26.5 × 115 = 3051 cu. ft. per hour, or 51 cu. ft. per minute.

Compare with this result an example of performance reported in *Power* for Nov. 22, 1910, page 2066. A Wheeler jet condenser, similar in type to Fig. 394 and installed with a 2000-kw. turbine, received cooling water at 51.0 deg. and discharged it at 90.4 deg., maintaining a vacuum of 28.50 in. or a condenser pressure of 0.73 lb., to which corresponds a steam temperature of 91.5 deg. From assumed values of steam rate and thermal data, it was calculated that about 182 cu. ft. of cooling water entered the condenser per minute. At 2 per cent, the free air from this water is 3.6 cu. ft., and a fair allowance would make the probable whole amount 6 cu. ft. per minute. It was estimated that the air and vapor mixture might have a temperature of 65 deg. and an absolute pressure of 0.60 lb. in the cylinder of the air pump during suction; for which conditions Fig. 391 gives a relative volume of about 50 cu. ft. Then the required air-pump capacity seems to be 300 cu. ft. per minute, while the actual displacement was 555 cu. ft. A volumetric efficiency of 0.9 would reduce the latter to 500 cu. ft. effective, but there remains a most decided discrepancy between 300 and 500, throwing doubt on the assumptions as to free-air quantity and as to realized cooling of the air and vapor mixture. In regard to the last condition, we note that a rise of temperature from 65 deg. to 74 deg. would bring the required volume above 500 cu. ft. per minute.

(*g*) Countercurrent Jet Condensers. — With any reasonable proportioning of the internal passages for steam, pressure will be almost uniform throughout the enclosed space of the condenser. A similar uniformity of temperature by no means follows, and the last example has supplemented Art. (*e*) in showing how advantageous is the scheme of cooling the gas-and-vapor mixture before sending it to the air pump. Two typical designs in which local cooling is effected in condensers of the mixing class are illustrated in Figs. 393 and 394.

The first thing to be noted in Fig. 393 is the thorough spraying of the cooling water and the low position of the exhaust inlet. The fine spray of water is heated nearly up to the temperature of the steam through which it falls, thus insuring a maximum absorption of heat; while the "air" rising from the condensed steam is cooled and dried by unheated spray, before passing through pipes to the top of the condenser. From this opposite flow of air and of water is derived the name "countercurrent." Meanwhile, the air in the injection water has nearly all been released by simple lowering of pressure, and escapes without being heated at all. Above the condenser is a vertical length of large pipe which serves as a separator, or a quiet chamber where drops of water can separate from the air current (if entrained) before the latter goes to the dry air pump. Special features of this design are, the equalizing pipe and the overflow pipe. The first gives the descend-

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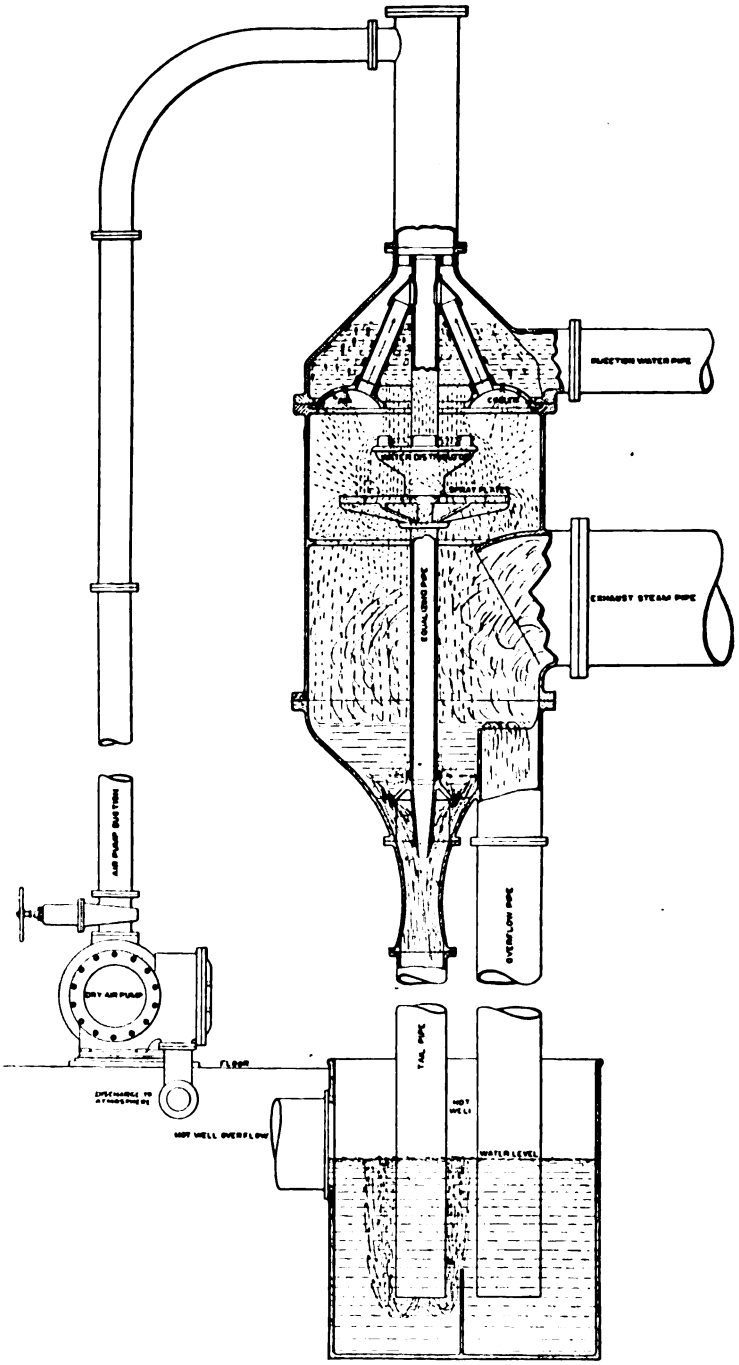

FIG. 393. — Large Barometric Condenser, illustration from bulletin of Allis-Chalmers Company.

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ing water current a chance to exert all the air entrainment of which it is capable, thus largely diminishing the service required of the air pump. The second insures the effectiveness of this ejector action: the tail pipe is so proportioned that it will be filled even by the smallest rate of water flow likely to occur, while at heavy loads all excess water goes down the large overflow pipe.

In Fig. 394 is seen the working out, with a different form of the condenser and its parts, of the same ideas of thorough spraying, of cooling the air from the exhaust steam by unheated spray, and of allowing the air in the injection water to escape at once into cool space. This water is distributed by seven parallel troughs, over the edges of which it falls in thin streams, and splashes on the spray plates. The condenser is placed right beneath the turbine, making a very compact

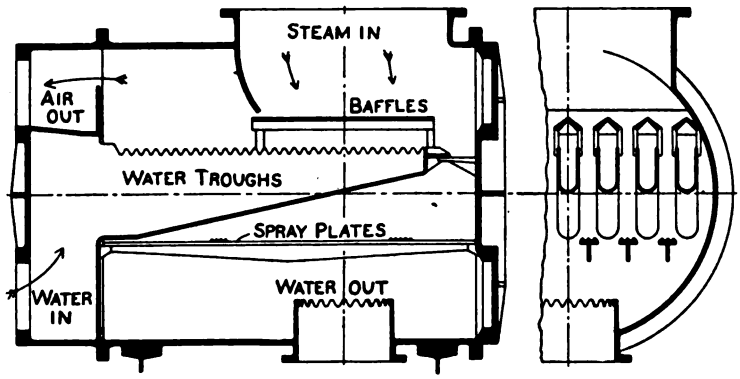

FIG. 394. — Tomlinson Mixing Condenser, Allis-Chalmers Company. See *Power* for Mar. 22, 1910.

unit. It is served by two centrifugal pumps, on the same shaft, of which the larger draws water from the condenser, the smaller sends a stream to the air ejector shown in Fig. 401.

Similar progressive cooling of the air pump "burden" toward exit is arranged for in modern surface condensers: but before enlarging upon this point, it will be well to take up the subject of heat transmission through tube walls.

(*h*) ACTION OF COOLING SURFACE. — The simplest case will be considered first, and elements of complication and uncertainty introduced afterward. Let

t_s = temperature of steam in condenser, now assumed to be uniform throughout whole space, or over all cooling surface.

t = temperature of cooling water, rising from t_1 at entrance to t_2 at exit.

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A = area of whole cooling surface of condenser, in square feet.
K = rate of heat transmission, in B.t.u. per hour per square foot of surface and per degree of temperature difference between substances on opposite sides of tube wall.
Q = heat transferred per square foot per hour.
Q_m = average transfer per square foot, so that AQ_m is the whole heat given up by the steam and absorbed by the water.
S = steam condensed per hour, pounds.
W = water used per hour, pounds.
$w = W \div S$ = cooling water per pound of steam.

We now assume K to be independent of the temperature difference and constant for all the surface involved. If the water temperature be averaged as t_m, on a base of surface traversed or of length of water path, an equation between heat transferred and heat absorbed by water from t_1 to t_2 gives,

$$AQ_m = AK(t_s - t_m) = W(t_2 - t_1). \quad . \quad . \quad . \quad (249)$$

The first question that arises concerns the value of t_m, or the manner of variation of water temperature along the path of flow. For an element of surface at a particular temperature t, we have the equation

$$Q\,dA = K(t_s - t)\,dA = W\,dt; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (250)$$

whence

$$A = \frac{W}{K}\int_{t_1}^{t_2} \frac{dt}{(t_s - t)} = \frac{W}{K}\log_e \frac{t_s - t_1}{t_s - t_2}. \quad . \quad . \quad . \quad . \quad (251)$$

Combining Eqs. (249) and (251),

$$\frac{t_2 - t_1}{t_s - t_m} = \log_e \frac{t_s - t_1}{t_s - t_2},$$

and

$$(t_s - t_m) = \frac{t_2 - t_1}{\log_e \dfrac{t_s - t_1}{t_s - t_2}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (252)$$

In Fig. 395, the full-line curves belong to this simple case, for the assumed temperatures $t_s = 100$, $t_1 = 70$, $t_2 = 90$ deg. The base is area A, with unity representing the whole surface traversed. Line DE shows the constant steam temperature t_s, curve ABC the variant water temperature t. Eq. (251) may be written,

$$A = C\log_e \frac{t_s - t_1}{t_s - t_2}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (253)$$

then if $A = 1.00$, the constant C becomes, with the temperature limits just named,

$$C = 1 \div \log_e 3 = 1 \div 1.0986 = 0.910.$$

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If the integration be carried from t_1 to some t (not to t_2), the abscissa of a point on curve ABC is found from

$$A = 0.91 \log_e \frac{30}{100 - t}.$$

If $t = 80$, for instance, $A = 0.91 \log_e 1.5 = 0.91 \times 0.4055 = 0.369$, at B in Fig. 395.

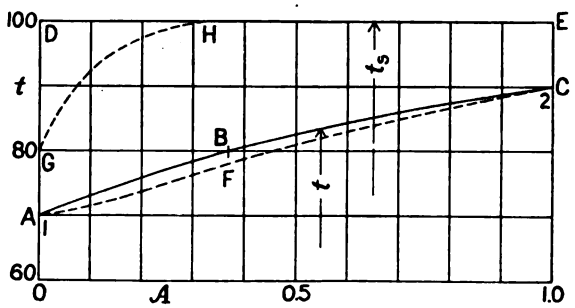

FIG. 395. — Curves of Temperature Variation.

With an arithmetic mean of t_1 and t_2, the average range in this example would be 20 deg.; but by substituting in Eq. (252) we find the true mean to be

$$(t_s - t_m) = \frac{20}{\log_e 3} = \frac{20}{1.0986} = 18.2 \text{ deg.},$$

or the mean ordinate of curve ABC is 81.8 deg. fahr.

(*i*) COEFFICIENT OF HEAT TRANSFER. — In a paper on "The Transmission of Heat in Surface Condensation," by Mr. G. A. Orrok, in Jour. A.S.M.E. for Nov. 1910, Vol. 32, will be found an account of some very extensive experiments made by the writer of the paper, with a reproduction, combination, and discussion of previous data. His conclusions are summed up in the expression (with changes of symbol),

$$K = C \frac{BR^5 M \sqrt{V_w}}{(t_s - t_m)^{\frac{1}{3}}}; \quad \ldots \ldots \quad (254)$$

where

K = average transfer coefficient, in terms described above.

C = a constant, given as 630.

B = coefficient of cleanliness of tube surface, ranging from 1.0 to 0.5.

R = ratio of steam richness, p_s/p in terms of Art. (*e*); it is the ordinate above a curve in Fig. 392, and ranges from 1.0 to perhaps as little as 0.2 near the air-pump outlet. Note what a tremendous influence this factor is given.

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M = material coefficient, ranging from 1.00 with clean copper down to 0.55 with badly-corroded brass.

V_w = velocity of water through tubes, in feet per second, ranging from 1 to 11.5 in the velocity experiments: about 8 ft. is recommended for normal or rated working, in a paper by the same writer in *Power* for June 1, 1909, page 966.

t_s and t_m have the same meaning as in the last article.

According to Mr. Orrok's experiments, the coefficient K is inversely proportional to the one-eighth power of the temperature difference, or the rate of transfer Q varies as the seven-eighths power of the difference. He considers it quite close enough to calculate the mean range by Eq. (252), then use this mean in Eq. (254) and in getting Q from K.

Continuing the conditions in the example under Fig. 395, which represent usual specification requirements (the most severe conditions) for turbine condensers, taking $B = 0.75$, $R = 0.97$, $M = 0.8$, $V_w = 8$, we find,

$$K = \frac{630 \times 0.75 \times 0.97^5 \times 8}{18.2} = 798.5.$$

And 800 for K, with a mean difference of 18 deg., gives $Q = 14{,}400$ B.t.u. per sq. ft. of condenser surface per hour. Turning to Table 20, and noting that input Q (as there given) is above condenser temperature and that the subtraction of output W from Q gives but a little more heat than goes to the condenser above its own temperature, we see that the cooling water must take up from 950 to 1050 B.t.u. per pound of exhaust, or say 1000 B.t.u. as a general average. Then the rate just deduced would call for a little more than 1 sq. ft. of surface per kilowatt or about 0.7 sq. ft. per horse-power. Twice this amount of surface has generally been considered a close allowance, but the tendency is toward a more definite control of the internal operation of the condenser, with resulting higher efficiency.

(*j*) THE PERFORMANCE OF SURFACE CONDENSERS is well represented by Table 26. Approaching this member of the plant from without, the prominent quantities are those given in columns 2 and 5. Seven pounds per square foot per hour seems to be a good average for the condensation rate, rising to ten in Nos. 11 to 15; the very high rates in Nos. 2 to 5 belong to experimental apparatus.

The transfer rate Q, equal to $w(t_2 - t_1)$ and to $K(t_s - t_m)$, is not in itself alone a good criterion of efficiency, but must be analyzed into its factors. With an unlimited supply of cold water, so that the difference $(t_s - t_m)$ is made large by having the range $(t_2 - t_1)$ short and low down

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on the thermometer scale, the condenser deserves no great credit for a high value of Q: but a good vacuum with but a moderate supply of comparatively warm water can be maintained only by high efficiency in heat transmission and in air removal. Good working along both these lines is most effectively shown by a small value of the difference $(t_s - t_2)$, in column 11, coupled with a high value of the transfer coefficient K.

An ideal condenser would raise the outgoing water to the temperature of steam saturation for the exhaust pressure p_0, making this pressure (or the realized vacuum) depend wholly upon the initial temperature t_1 and the quantity ratio w of this water. A large difference between t_s and t_2 generally means poor conduction of heat, due to foulness of tubes or to excessive presence of air in the condenser. In extreme cases, there may be so much air that even near the exhaust inlet the existing temperature t in the steam space will be appreciably less than t_s for p_0. The influence of air-pump capacity is strikingly shown in tests 2 and 3 with turbine No. 9, by the decrease of $(t_s - t_2)$ with rise of p_0 and consequent shrinkage in the volume of air to be handled; displacement remaining (presumably) nearly the same, the pump is far more effective at the higher pressure. Unusually complete air removal is probably the chief reason for the high efficiency, in heat transfer and in vacuum maintenance, of the type of condenser represented by Nos. 11 to 15 in Table 26.

The difference $(t_s - t_w)$ in column 12, or the drop from exhaust steam to water of condensation, is of some economic interest when the condensate is to be used as boiler feed. Chiefly, its high or low value is a characteristic of different types of air-pump arrangement. With a plain wet vacuum pump, handling water and air together, t_w must be lowered well toward t_1, to keep down the volume of uncondensed vapor. With a separate dry-air pump, the liquid condensate may be kept warmer, by diverting it from the coolest tubes near the bottom of the condenser, and using these to chill and dry the air and vapor mixture. A negative value of this difference, physically an impossibility, throws doubt on the accuracy of pressure measurement.

The ideas in the last two paragraphs apply equally to the jet condenser, and with some changes of form these statements might have been added to Art. (g). Ideal action would give the water current (mixed cooling water and condensed steam) the temperature t_s of the entering exhaust. With combined evacuation, as in Figs. 387 and 388, the whole mixture must be cooled well below t_s, to diminish the vapor volume. But with the principle embodied in Figs. 393 and 394, the water current may be kept almost as warm as the steam — witness the performance quoted in Example 57.

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TABLE 26 PERFORMANCE OF

No.	Old No.	1 Total Cooling Surface, Sq. Ft. A	2 Steam per Sq. Ft. per Hour. s	3 Water per Lb. of Steam. w	4 Rise of Temperature of Cooling Water. t_2-t_1	5 Heat per Sq. Ft. per Hour. Q	6 Condenser Pressure, Lb. Absolute. p_0	7 Steam Temperature at p_0. t_s	8 Initial Temperature of Water. t_1	9 Final Temperature of Water. t_2	10 Temperature of Condensed Steam. t_w
1.1	953	170	4.19	64.1	15.5	4160	0.68	89.2	51.0	66.5	71.9
2	958	"	6.47	41.7	22.6	6100	0.98	101.2	50.6	73.2	94.1
3	963	"	9.58	28.2	34.2	9230	1.65	119.2	50.1	84.3	117.8
4	968	"	10.94	24.8	39.3	10660	2.07	127.4	50.0	89.3	128.4
2.1	987	101	17.22	31.8	30.0	16440	0.67	88.9	45.3	75.3	90.0
2	991	"	18.81	13.3	73.0	18250	1.95	125.4	45.7	118.7	128.9
3.1	1014	101	12.52	45.9	22.0	12640	0.51	80.2	46.0	68.0	76.3
2	1035	"	18.30	14.2	71.5	18560	1.71	120.5	42.5	114.0	121.9
4.1	1055	62	35.6	26.0	35.9	33220	0.96	100.4	41.3	77.2	101.3
2	1061	"	27.8	14.4	66.3	26530	2.16	129.0	41.7	108.0	132.0
5.1	8	307	7.89	30.2	34.0	8090	0.70	90.0	50.4	84.4	65.7
2	6	"	7.91	49.8	20.8	8210	0.52	81.9	50.4	71.2	54.0
3	3	"	13.12	24.0	43.7	13780	1.00	102.0	50.4	94.1	79.5
4	1	"	13.03	51.4	20.8	13930	0.56	82.3	50.4	71.2	58.5
6.1	4–7	962	7.25	20.2	54.4	7960	1.67	119.8	50.5	104.9	61.2
2	4–8	"	7.13	24.9	43.4	7700	0.88	97.9	50.5	93.9	71.4
3	4–12	"	7.14	39.2	27.3	7620	0.56	82.3	50.5	77.8	64.4
4	411	"	7.42	36.9	29.0	7930	1.06	104.6	71.9	100.9	87.3
7.1	6–6	959	6.15	40.4	26.1	6480	0.78	93.6	50.8	76.9	71.1
2	6–1	"	7.99	32.6	32.9	8580	0.77	93.2	50.8	83.7	65.9
8	101	1883	6.80	98.0	10.8	7200	0.52	80.6	49.1	59.9	68.9
9.1	58	25000	3.26	153.0	6.3	3140	0.48	78.4	33.2	39.5	53.9
2	54	"	7.06	77.1	14.3	7780	0.46	77.1	33.5	47.8	57.3
3	14	"	7.05	78.6	13.6	7530	0.80	94.5	73.0	86.6	86.4
4	38	"	9.50	53.5	19.6	9950	0.74	92.0	37.7	57.3	71.2
10.1	I	4520	3.52	71.3	14.6	3660	0.44	75.8	59.5	74.1	69.8
2	II	"	4.48	52.7	19.1	4510	0.53	81.4	60.6	79.7	82.6
3	III	"	6.25	54.6	19.4	6620	0.67	88.8	57.6	77.0	81.0
11	1044	2205	11.2	70	14.3	11220	0.38	71.5	52.4	66.7	68.8
12	1007	3200	11.0	77	13.0	11000	0.82	95.3	76.5	89.5	89.8
13	1008	3350	10.4	71	14.0	10400	0.79	94.1	72.8	86.8	87.0
14	1102	3350	11.5	80	12.5	11540	0.88	97.6	82.4	94.9	96.5
15	991	1800	8.8	54	18.5	8800	1.58	117.7	92.8	111.3	110.1

Columns 2 to 5 are tied together by the relation $Q = sw\ (t_2 - t_1)$.
Columns 6 to 10 are observed quantities, except that steam temperature t_s is taken from the table, for p_0.
The meaning of the temperature differences in columns 11 and 12 is explained in Art. (*j*).
The factor in column 13 is used in getting the mean range $(t_s - t_m)$, according to Eq. (252).
In column 15, K is the quotient $[Q \div (t_s - t_m)]$.

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SURFACE CONDENSERS. TABLE 26

11 Difference, Steam and Warm Water. $t_s - t_2$	12 Difference, Steam and Hot Well. $t_s - t_w$	13 $\frac{t_s - t_1}{\log_e \frac{t_s - t_1}{t_s - t_2}}$	14 Mean Difference of Temperature. $t_s - t_m$	15 B.t.u. per Sq. Ft. per Deg. per Hour. K	16 Water Velocity, Ft. per Sec. V_w	References, Notes, etc.
22.7	17.3	0.521	29.8	114	0.8	From paper by Prof. R. L. Weighton, Trans. Inst. Nav. Arch., 1906, Vol. 48, 122–156. Experiments at Armstrong College, Glasgow. No. 1, old-type plain condenser, ¾ in. tubes, 4 ft. long, 5 passes. No. 2, " contraflo " condenser about as in Fig. 397, ⅝ in. tubes, 4 ft. long, 4 passes. No. 3, same as No. 2, but with separate dry air pump. No. 4, like No. 2, but with tube length shortened to 2.5 ft.
28.0	7.1	0.592	38.2	160	0.8	
34.9	1.4	0.683	50.1	181	0.8	
38.1	−1.0	0.709	55.4	193	0.8	
13.6	−1.1	1.166	25.7	640	4.5	
6.7	−3.5	2.476	29.5	618	2.1	
12.2	3.9	1.037	21.2	597	4.6	
6.5	−1.4	2.485	28.8	644	2.0	
23.2	−0.9	0.932	38.5	864	4.6	
21.0	−3.0	1.426	46.6	570	2.0	
5.4	24.3	1.966	17.4	465	1.3	From paper by Prof. E. Josse, *Zeit. Ver. deutsch. Ing.*, 1909 I, Vol. 53. No. 5, special experimental condenser, water channel of changing cross section, 13.93 ft. long, tubes 0.59 in. diam. No. 6, tubes 0.71 in. diam., 2 passes, length 15.1 ft. No. 7, first test, plain tubes; second test, " spiral " (helical) baffle strips in tubes. No. 8, condenser in regular service.
10.7	27.9	1.079	19.3	425	2.2	
7.9	22.5	1.878	23.3	591	1.7	
10.1	23.8	1.061	19.6	710	3.2	
14.9	58.6	1.953	35.4	223	0.7	
4.0	26.5	2.281	17.5	439	0.8	
4.5	17.9	2.470	14.0	555	1.3	
3.3	16.9	1.535	12.7	625	1.2	
16.7	22.5	0.940	27.7	235		
9.5	27.3	1.497	22.0	390		
20.7	11.7	0.419	25.8	279		
38.9	24.5	0.270	41.7	75	4.2	Turbine-base condenser Fifty-ninth Street Power House, New York: with turbine No. 50 in Table 20 and engine No. 27 in Table 13.
29.3	19.8	0.399	35.6	218	4.6	
7.9	8.1	1.044	13.1	575	4.7	
34.7	20.8	0.445	43.8	227	4.3	
1.7	6.0	2.261	6.4	568		Balcke condenser, STODOLA IV, 551, 556; air and vapor guided over coldest tubes before going to air pump.
1.7	−1.2	2.503	7.6	592		
11.8	7.8	0.971	20.0	331		
4.8	2.7	1.381	10.3	1090	8.6	Condensers with Parsons vacuum augmenter: from table of data presented by Mr. R. J. Walker, in discussion of paper by Mr. D. B. Morison, Trans. Inst. Nav. Arch., 1908, Vol. 50, page 169. In these tests the steam rate was not given, but is derived from heat absorbed Q, by assuming 1000 B.t.u. from the pound of steam.
5.8	5.5	1.177	11.1	1090	7.1	
7.3	7.1	1.072	13.1	790	6.2	
2.7	1.2	1.728	7.2	1590	7.4	
6.4	7.6	1.358	13.6	645	6.7	

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(*k*) EFFICIENCY OF COOLING SURFACE. — Some brief discussion of the influences given place in Eq. (254), partly a review and summary, seems appropriate. Concerning cleanliness and material, as bearing on conductivity, it is enough to say that the condition of the surface — whether clean metal or coated with an insulating film or sheathing — is far more important than the metallic composition of the body of the tube: in other words, for the metallic wall, surface resistance is far greater and more widely variable than internal resistance. As between the two types of steam machine, the turbine has the advantage over the engine that it does not send heavy oil into the condenser, likely to coat the tubes with grease on the steam side.

That water velocity has a considerable influence is readily accounted for by the fact that water is a poor conductor, and must be heated by convection. The tendency is to form a thin layer of warm water about a cooler core, within the tube. Rapid flow, or the insertion of baffles as in No. 7 of Table 26, keeps the current well stirred up, continually bringing fresh portions into contact with the walls. Also, with dirty water, swift flow scours out the tubes and keeps them clean. A limit to economical velocity is imposed by the rapid rise of resistance and of work by the circulating pump, since the pressure required varies as the square of velocity.

The poor conductivity of a layer of water accounts for the disadvantage of "water drowning": in a large condenser, the heavy shower from above covers the lower tubes so thickly that steam has poor access to them. The scheme of partial drainage, illustrated in Figs. 396 and 397, is intended to overcome this difficulty and keep the tubes comparatively dry.

The deadening effect of air drowning has been alluded to in the last article, and in Fig. 395 is tentatively represented by the dotted curves AFC and GH. In the active space near the steam inlet, flow of steam is so strong that air cannot accumulate; but the cooler tubes toward the air-pump outlet will be surrounded by a mixture of increasing air richness. To show the consequent low conduction, curve AFC is made to rise very slowly at first; and while the cooling of the air mixture does not involve much heat abstraction, the drop from H to G is spread over a considerable length of base. The curves are wholly illustrative, not being based on any quantitative investigation. The practical inference, from the viewpoint of design, is that a relatively large amount of surface will be required for the operation of cooling and drying the air-and-vapor mixture.

(*l*) VARIOUS SURFACE CONDENSERS. — Fig. 396 illustrates the introduction of drainage plates or "rain plates," slightly pitched from

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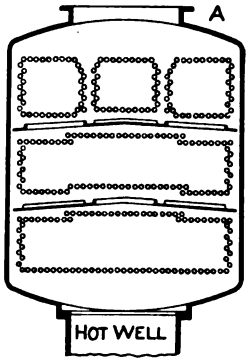
the middle toward the walls, down which the water runs instead of falling on the tubes below. As shown in the plan at B, large holes through the plates give passage to steam, and these holes are " staggered " in sequence, so as to promote circulation. Evidently, this arrangement will largely prevent cooling of the water of condensation. Note the passages formed between three groups of tubes in the first section; these



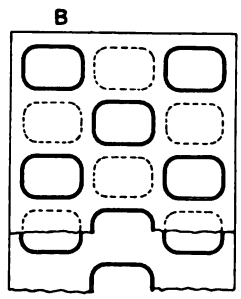


FIG. 396. — Wheeler Dry-tube Condenser.

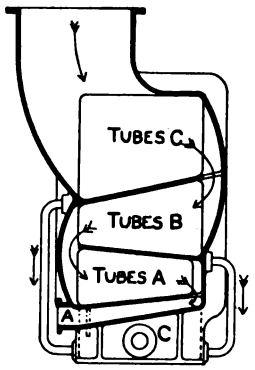


FIG. 397. — Early Form of Contraflo Condenser.

give better access of steam to the second section, and insure high activity in the latter. Some such scheme of permitting freer flow among the higher tubes is now quite common.

The pioneer design * embodying the idea of partial drainage, with the double purpose of not flooding the lower tubes and of delivering warm water, is shown in Fig. 397. The free spaces at the sides, between tube sections and formed by swelling out the sides of the shell, are intended to promote uniform, parallel flow of steam. Note the very distinct separation of air and water; the latter goes out at A, after thorough cooling in the lowest tube section, while the water runs into a small reservoir at one end (from sections B and C through the pipes shown), whence it is withdrawn at C. This was an experimental condenser: in working designs the shell is of simple cross section, and inclined baffle or guide plates are interposed among the tubes, so as to give partial drainage and to make the steam current flow in a definite channel of rapidly decreasing cross section.

Figure 398 shows a device of which the effectiveness has been fully shown in the latter part of Table 26. A steam jet S exerts a strong

* That of Weighton; see reference from Nos. 1 to 4 in Table 26. The name "contraflo," used as a trade name, is not based on countercurrent action, but on a special effort to spread the steam current sidewise at entrance, and then have it flow squarely across and among the tubes.

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ejector action at D, drawing air and vapor from the main condenser. The small auxiliary condenser cools the ejector discharge, liquefying all excess of steam, and delivers it to the air-pump suction at F. With the increases of pressure represented by the hydraulic head h, the mixed condensate is easily handled by pump of moderate capacity. The

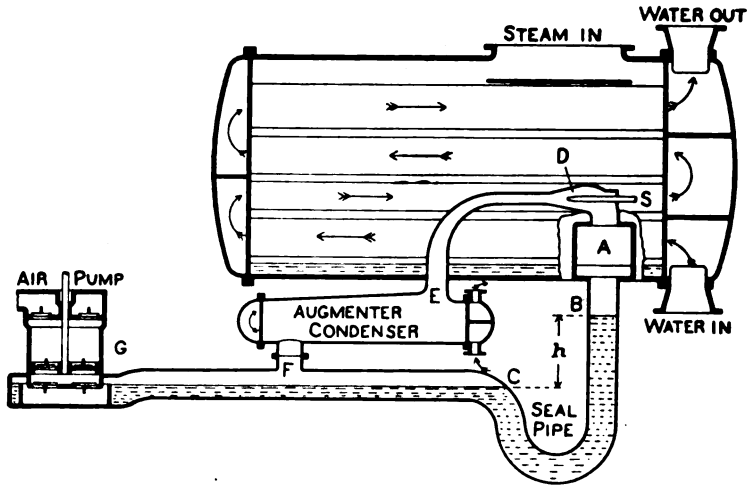

FIG. 398. — The Parsons Vacuum Augmenter.

steam consumed by the ejector is said to be about one per cent of the normal consumption of the turbine.

As more or less special types may be mentioned the following:

The feed-heater condenser, in which the boiler feed is passed through a top section of tubes, above those through which cooling water is pumped.

The water-works condenser, with steam inside of the tubes and water outside, in order to get a smaller resistance to flow of water. This is placed in a bypass on the suction pipe of the pumping engine, and the needed diversion of water is secured and varied by a butterfly valve in the main pipe.

The "countercurrent" condenser, with steam entering at the bottom and cooling water at the top. The condensation water is fully warmed by showering down through entering steam, while the dry-air pump draws from the top of the condenser. Since air is heavier than steam, and cooling causes a further increase of density, this scheme is somewhat illogical.

It may be remarked that terms describing relative directions of flow have a conventional and limited meaning when used in connection

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with the condenser. All jet condensers have parallel flow of cooling water and condensed steam, all surface condensers opposite flow of steam and water; so that "countercurrent" and its implied opposite refer to the direction of air flow.

(*m*) AIR PUMPS. — Typical simple, combined-service or wet vacuum pumps are shown in Figs. 387 and 398, the latter not specifically belonging to the condenser which it accompanies. The vertical pump, single acting, has three sets of valves, while the horizontal pump has but two sets. In the latter the working "piston" is really a water surface, which rises and falls in the chambers H, H. Air-pump valves are of the flat disc form in Figs. 387 and 399, not cone-seated as sketched in Fig. 398.

An improved vertical pump with but one set of valves, or "suction valveless," is shown in Fig. 399. A certain amount of water always remains in the bottom of the cylinder, and more flows in while the piston rises and during most of the down stroke. The lower face of the piston P is conical, quite closely fitting the cylinder bottom; then as it comes down the standing water is forced out, impelled around the curved surface W, W, and shot in through the ports. This spray of water will exert some injector action upon the air present, helping to fill the suction space. The pump is, of course, crank driven, since the stroke of the piston must be very definite.

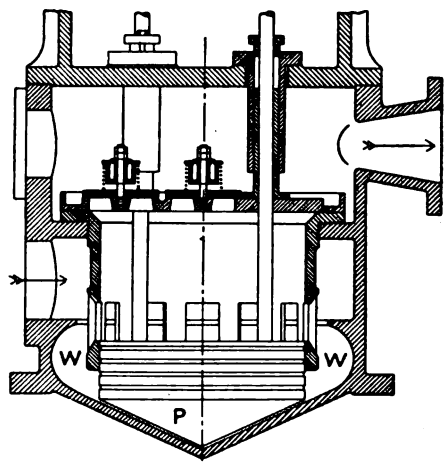

FIG. 399. — The Edwards Air Pump.

As already remarked, the dry-air pump is, in form and working, very much on the lines of the common air compressor. Its function is to receive a mixture of air and steam at from 0.5 to 1.5 lb. absolute and compress it to atmospheric pressure. Often the cylinder is water-jacketed, since so great a range of compression involves a good deal of heating; and sometimes there are two stages of compression, in successive cylinders. The inlet valves are commonly operated mechanically, to save suction vacuum.

(*n*) CLEARANCE AND VOLUMETRIC EFFICIENCY. — The purpose of the air pump, of either class, is to remove as large a volume as possible of the air and vapor coming to it; and the greatest obstacle in the way of

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efficiency is clearance — using this term in the same sense as with the engine cylinder. Suppose, for example, that a dry-air pump has a pressure range from 0.75 lb. to 15 lb., or a ratio of 1 to 20. Under the law $pv = C$, the compressed volume would be 0.05 of the suction volume; but really the compression will be quite nearly adiabatic, or enough so to increase this final volume to 0.09 or 0.10. If the clearance is, say, 3 per cent of the displacement, only 0.65 to 0.70 of the charge will be expelled, the remainder expanding behind the piston as it returns, spoiling the suction vacuum of the pump or delaying the indraft of fresh charge. The volumetric efficiency, or the ratio of actual suction volume to displacement, will be only 65 to 70 per cent.

In wet-air pumps, the effect of clearance is largely neutralized by the presence of water. In the three-deck vertical pump, like Fig. 398, the piston will always carry enough water above it to fill the clearance space at the top of the cylinder, so that all air will be expelled. During the down stroke pressure is equalized on the two sides of the piston, barring valve resistance, so that even if the water above the piston has received heat of compression, its vapor tension cannot diminish volume capacity. On the last point the single-acting pump has an advantage over all those in which the same side of the piston, or the same moving surface, is active in both suction and compression. Even though all air be expelled from the pumps of Figs. 387 and 399, vapor from the water will fill or tend to fill the suction space, and will have to be condensed or compressed by the entering charge. The spray injected by the piston action of the Edwards pump will be more active than a plain water surface in thus condensing vapor.

The introduction of water to fill the clearance space of a dry-air pump would be a very poor expedient, because this water would be warmed during compression and would have a high vapor tension. The best scheme is to use a "snifting" valve, which controls a passage connecting the two ends of the cylinder, and is opened (mechanically) soon after the crank passes each dead center and kept open for a short time. Then the high-pressure clearance content at one end passes into the other end, where pressure is low because compression has just begun. With this arrangement, the volumetric efficiency may rise as high as 95 per cent.

(*o*) WATER EJECTORS. — In Fig. 400, the point of special interest is the air pump. As to the condenser in general, water enters at A and is sprayed downward from an annular distributor C. The whole product of mixing and condensation, in a stream filling the large funnel or nozzle, sweeps into space E, where the air separates and is drawn off through pipe K. A plain centrifugal pump takes water from the con-

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denser; and on the same shaft with it is the impeller of turbine pump P, which sends a stream of cool water into the ejector nozzle. The stream is delivered in layers, or broken into spray, and the interstices among drops being necessarily filled with their surrounding atmosphere, this

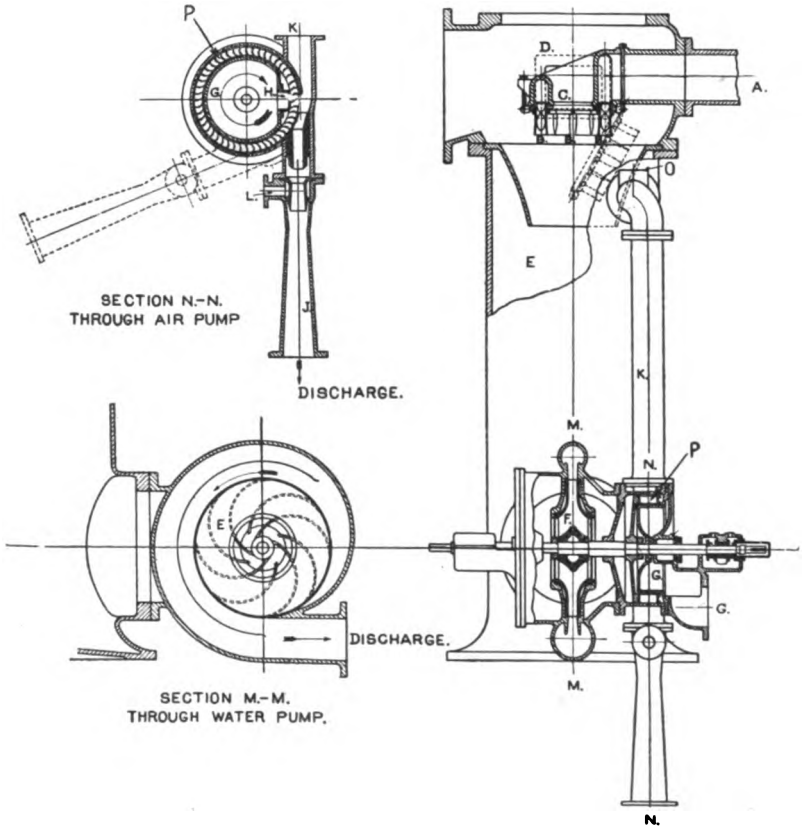

FIG. 400. — The Leblanc Condenser and Pumps. From Bulletin of Westinghouse Machine Company.

air from the condenser is carried out by and with the water. Partly by suction of the vacuum, partly by the impeller, the current is given sufficient velocity to expel it against outside pressure. If the cold water tank is not high enough for flow to the pump by gravity, steam is let in at L when starting up. This air pump can equally well be used with a surface condenser.

A fundamentally equivalent device, although quite different in form, is shown in Fig. 401. Water, from a special centrifugal pump,

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comes in at W and is discharged through a ring nozzle in a hollow conical jet or sheet. Air from A has parallel admission both inside and outside of the cone of water. The latter breaks into spray, yet with enough velocity for self-expulsion, and air is mixed with and carried out by this broken stream.

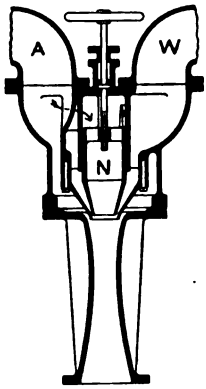

FIG. 401. — Tomlinson Air Ejector, with condenser in Fig. 394

If water is not plentiful, the same supply may be used over and over again in these air pumps. The scheme is essentially that of the old ejector condenser, but with the advantage that the whole condensate need not be cooled to the low temperature necessary for effective air removal at high vacuum. Devices of this class are not subjected to the definite volumetric limitations of piston pumps.

(*p*) POWER CONSUMED BY CONDENSER PUMPS. — With steam-driven pumps, this is a rather indeterminable quantity; and if the exhaust from these auxiliaries be used to heat the feed water, their power cost is almost zero, as shown in § 26 (*g*), page 235. With motor drive for the pumps, the power consumed is very easily measured, and is to be deducted from gross output. Of the turbines in Table 20, we have the record that the 4000-kw. unit listed as No. 35 required 60 kw., or 1.5 per cent of rated power for the condenser pumps: and the 1500 kw. condenser plant, No. 10 in table 26, consumed 19 kw. in the same service. The average range in large and well-kept plants is probably from 1.3 to 2 per cent of the rated output, varying hardly at all with degree of loading.

(*q*) THE COOLING TOWER. — When but a scanty supply of water is available, some plan of artificial cooling and repeated use must be resorted to. This cooling is effected by exposing a large surface of warm water to a current of air and thus encouraging vaporization, the latent heat for which will come from the water itself. The weight of vapor formed and carried off by the air is about equal to that of the steam condensed, which relation determines the amount of make-up water required. The simplest scheme is to use a large pond, if at hand, discharging warm water into one end, drawing cool water from the other; the first advance upon this is to spray the water upward in fountain jets when discharging into the pond; but under common space limitations a cooling tower must be added to the plant.

This tower is a circular or rectangular shell of light plate — in effect, a chimney stack much shortened vertically (20 to 40 ft. high) and

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very much enlarged laterally. At the top is a set of distributing troughs, to which the water from the condenser must be pumped; from these it trickles down over "mats" made of wooden slats or of woven wire screens, which fill the space within the tower. Sometimes natural "draft" is depended upon for air circulation, but oftener fan blowers, of the helical or screw-propeller type, are installed. Of course, the capacity of a cooling tower is most severely taxed in warm and damp weather, when the air is so nearly loaded (saturated) with natural moisture that it can take up little more. For a good description and discussion of this apparatus, see paper by Mr. J. R. Bibbins on "Cooling Towers for Steam and Gas-power Plants," in Trans. A.S.M.E., 1909, Vol. 31, 725.

A combination of the cooling-tower idea with the surface condenser has been thought of and tried — see, for instance, Trans. A.S.M.E., Vol. 14, 690. This self-cooling surface condenser consists of a nest of pipes, with steam inside, water trickling down outside, and air blown over the water-coated surfaces. The scheme is workable, but efficiency of heat transfer is low, hence a large amount of cooling surface must be provided; and the construction is necessarily such that both cost and space occupied per square foot of surface will be high.

EXAMPLE 58. — Suppose that a cooling tower plant, when handling water at the rate of 25 lb. per pound of steam condensed, returns it at 75 deg. fahr., and that the effective capacity of the air pump is 0.7 cu ft. per pound of steam. What vacuum may be expected?

Let us assume that the engine uses dry-saturated steam at 135 lb. abs. and requires 13.5 lb. of steam per horse-power-hour. The total heat of this steam is 1193 B.t.u., and the heat converted into work per pound is $2545 \div 13.5 = 196$ B.t.u. With a reasonable allowance for radiation, we may say that the total heat of the steam coming to the condenser will be about 975 B.t.u.

If the condenser is of the jet type, all of this heat above 75 deg., or $975 - 43 = 932$ B.t.u., will go to raise 26 lb. of water (cooling water plus condensed steam) to the final temperature t_2; then

$$t_2 - t_1 = t_2 - 75 = \frac{932}{26} = 34.8,$$

and $t_2 = 110$ deg. To this corresponds an absolute pressure of 1.27 lb., which is the lower limit of the attainable, so that the ideal vacuum is

$$29.92 \times \frac{14.7 - 1.27}{14.7} = 29.92 \times \frac{13.43}{14.7} = 27.3 \text{ in. mercury.}$$

Now 0.7 cu. ft. of displacement per pound of steam is equivalent to $0.7 \times 62.4 = 43.7$ cu. ft. per cubic foot of feed water. Let us assume further that the free-air volume of the air to be removed is equal to the volume of the feed water — compare Art. (*f*). In Fig. 391, on the ordinate at 110 deg., 43.7 cu. ft.

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interpolates at about 1.65 lb. pressure. This is then the pressure to be expected if the air and vapor mixture must be removed at the full temperature of 110 deg.

But suppose that by local cooling this mixture is reduced say to 90 deg. before it goes to the air pump; then on the 90-deg. ordinate of Fig. 391 a volume of 44 cu. ft. is found to correspond with a pressure of about 1.06 lb., while the volume at 1.27 lb. is only about 28 cu. ft. If the air-pump displacement is in excess of the volume of vapor-saturated air at the temperature existing, the extra space will be filled by excess of vapor, drawn from the water at that temperature. With as little as two-thirds of the displacement named, at the temperature named, ideal vacuum should be realized. With a surface condenser and an equivalent effectiveness of air cooling, the condenser pressure would depend upon the steam temperature t_s as related to the final water temperature t_2; on this point, refer to Table 26, and note that $(t_s - t_2)$ may be as little as 5 deg.

§ 54. The Rotary Engine

AN IMMENSE AMOUNT OF INVENTION and ingenuity has been expended upon the problem of devising a machine in which the static-pressure cycle could be applied to a rotating instead of a reciprocating working element. Inherent difficulties of the first magnitude have kept these efforts from success, while the advent of the turbine has satisfied the demand which they were intended to meet. A description of the various rotary engines that have been patented would belong rather to the study of mechanism than to practical steam engineering; and to illustrate the general principle of action we shall use but the one simple type outlined in Fig. 402.

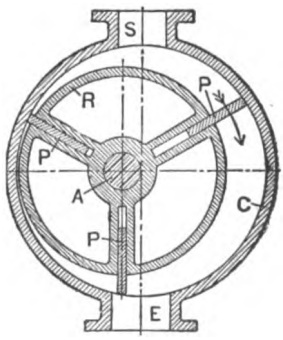

FIG. 402. — Outline of Typical Rotary Engine.

In this machine steam enters continually at S and escapes at E, only a plain throttle valve in the inlet pipe being needed to control the flow. Sliding vanes or pistons, P, P, P, project from the rotor R and touch the casing C, furnishing the moving surface upon which the working steam force acts. Some simple contrivance, such as a circular groove in the ends of the casing and projections on the vanes, must be added in order to keep the latter out against the casing.

This particular scheme has the fault that it makes absolutely no provision for expansion of steam below the inlet pressure, which puts it in a class with the small direct-acting steam pump. Better inventions have met this requirement to some degree, although far less effectively than the piston engine. But the great and determining

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drawback has been the inability to hold steam, or the tendency of the machine to excessive leakage. The circular piston, moving in its cylinder with a broad contact, is the simplest possible working element for the static-pressure cycle, and it is hard enough to keep the piston tight. The rectangular vane, with narrow contacts and variant rubbing speed at different points of its perimeter, cannot be made and kept steam-tight; and there is a further chance for leakage between the ends of rotor and casing. As regards thermal wastes, it is evident that a considerable portion of the metal surface which the steam touches is exposed alternately to high and low temperatures. Further, since the machine develops its power by the combination of a small driving force with a high velocity, the rubbing friction due to an attempt to keep the piston tight will absorb an excessive amount of power. These unavoidable bad features have completely overshadowed the advantages of compactness and freedom from shaking force.

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APPENDIX

TABLES OF THE PROPERTIES OF STEAM

A. List of Tables and Diagrams

TABLE I. TEMPERATURES FOR VARIOUS PRESSURES. — Since Table II has temperature for the argument, Table I gives the temperatures corresponding to equally spaced and numerically simple values of pressure: to be used before entering Table II, when pressure is the initial quantity. Pages 578 and 579.

TABLE II. PRINCIPAL STEAM TABLE, for the properties of saturated steam (columns 3, 4, and 5 for superheated steam). This table covers at one-degree intervals the range from 32 deg. to 550 deg. fahr., and extends at ten-degree intervals to the critical temperature, which is taken as 689 deg. Pages 580 to 601.

TABLE III. SUPPLEMENTARY STEAM TABLE. — This contains some regularly used reference columns in which ten-degree spacing is close enough, also certain columns which find their purpose in the illustration rather than the application of the properties of steam. Pages 602 and 603.

TABLE IV. TEMPERATURE FACTOR f_t. This extends into the high-temperature range the quantity given in column 5 of Table II, to be used in calculating the specific volume of superheated steam. Page 603.

TABLE V. PRESSURE FACTOR f_p. — This covers the range of high pressures, above the field of application of formula (65), § 12 (*i*). Page 603.

TABLE VI. SPECIFIC VOLUME OF SUPERHEATED STEAM. — Diagram: vertical base, pressure on a scale uniform for saturation temperature; principal scale for horizontal ordinate (inclined ruling), temperature fahrenheit; secondary scale (vertical ruling), degrees of superheat. The curves are lines of constant volume, and the horizontal lines of constant pressure are divided for equal volume increments. Pages 604 and 605.

TABLE VII. TOTAL HEAT OF SUPERHEATED STEAM. — Diagram: vertical base, pressure on a uniform scale; the horizontal ordinate is

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total heat h under constant pressure — see § 13 (i). Curves marked TT are lines of equal temperature or isothermals; those marked SS are lines of equal superheat. Pages 606 to 610.

TABLE VIII. ENTROPY OF SUPERHEATED STEAM. — Diagram: vertical base, temperature fahrenheit; horizontal ordinate, entropy n above 32 deg. fahr., acquired in the reception of total heat h. The constant-entropy lines are inclined, to economize space. Principal curves are for operation of heating under constant pressure, at the pressure corresponding to each ten-degree point of temperature on the saturation line: at top of diagram is a scale of corresponding pressures for these curves. The cross curves show equal superheats, at each fifty degrees above saturation. Pages 611 to 615.

B. Alphabetical List of Symbols for Steam Quantities,

WITH DEFINITIONS OF THE QUANTITIES AND REFERENCE TO EXPLANATIONS IN THE TEXT, NEARLY ALL IN CHAPTER III.

Steam-table headings in heavy type.

All volumetric and thermal quantities are per pound of substance, whether water, steam, or mixture.

A The heat equivalent of work, or the value of one foot-pound in B.t.u., equal to 1/778 or 0.001285; used in the external-work expression APv. See Eq. (3), page 35.

APu Table III, column 8. External work of vaporization; §13 (f).

APw Table III, column 7. External energy of the liquid; § 13 (g).

In general, if a volume-increase v (cu. ft.) takes place under a constant pressure p (lb. per sq. in.), the work done is 144 pv or Pv ft. lb., using P for pressure in pounds per square foot: then APv is the heat equivalent of this work, in B.t.u. When making calculations, it is best to use 144 Apv or 0.185 pv. In APu and APw, u and w are particular volume symbols, as defined below. See § 7 (d).

a Table II, column 11. Entropy of the liquid; } see § 13 (l),

b Table II, column 12. Entropy of vaporization; } also N below.

c Specific heat, in following particular values.

c_p Specific heat of superheated steam under constant pressure; § 13 (h) and Fig. 46.

c_v Specific heat of superheated steam at constant volume; § 13 (k).

c_w Table III, column 6. Specific heat of water; use also plain c; § 13 (a).

d Density, in pounds to the cubic foot.

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d_s Density of saturated steam, the reciprocal of the specific steam volume s; § 12 (*c*).

d_w Table III, column 3. Density of water at temperature t and pressure p as in columns 0 and 1, or weight of one cubic foot in pounds; § 12 (*c*).

f_p Table V. Pressure factor, to be used with f_t in calculating the specific volume of superheated steam, as explained in § 12 (*i*). This supersedes Eq. (65) at 900 lb. pressure.

f_t Table II, column 5, and Table IV. Temperature factor, to be used with f_p in calculating the specific volume of superheated steam; § 12 (*i*).

H Table II, column 8. Total heat of saturated steam, dry or of the quality 1.00, formed under a constant pressure p from water at 32 deg. fahr.; § 13 (*c*).

h Table VII. Total heat of superheated steam; § 13 (*i*) This is also used as a general symbol for the total heat (under constant pressure and above 32 deg.) of steam of any quality or condition — see § 13 (*d*).

h_s Heat added to the pound of steam in superheating; § 13 (*h*).

I Internal work or energy of one pound of steam, of any quality or condition; § 13 (*f*), (*g*) and (*j*).

K Table II, column 10. Internal or intrinsic energy of dry-saturated steam, above the state of water at 32 deg. fahr.; § 13 (*f*).

k Internal energy of the liquid, differing by but a very little from the heat of the liquid or q; § 13 (*f*).

l Table II, column 9. Inner latent heat, or the internal-energy portion of the whole heat r which is added during vaporization under a constant pressure p; § 13 (*f*).

m Proportion of moisture in a mixture of water and steam, or the weight of liquid in one pound of the mixture; the "fraction of moisture" or "degree of wetness."

N Table II, column 13. Total entropy of dry saturated steam, acquired in receiving the total heat H. It is the sum of the entropy of the liquid N_q (called *a*, column 11) received with the water heat q, and of the entropy of vaporization N_r (called *b*, column 12), received with the latent heat r. See § 13 (*l*).

n Table VIII. Like h, n is used as a general symbol, especially for the total entropy of superheated steam above the state of water at 32 deg. See § 14 (*b*), page 91.

n_s Used for the entropy acquired by the steam in being superheated at constant pressure, or in addition to the saturation value N. See § 13 (*l*).

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P Pressure in pounds per square foot, as in APu, etc., above; see § 7 (*d*).

p Table I, argument, Table II, column 1. Pressure of steam in pounds per square inch, absolute or above the zero of perfect vacuum; § 12 (*a*).

ps Table III, column 5. The product of pressure p and specific steam volume s.

pw Table III, column 4. The product of pressure p and specific water volume w.

These quantities are useful in investigating the properties of steam and comparing it with a perfect gas — see § 12 (*e*). Also, they lead to APu and APw, noting that $pu = ps - pw$: see APu, etc., above.

Q Heat of formation of the pound of steam, whether wet, dry-saturated, or superheated. This heat received from the fire is of the same nature as total heat H or h, but is measured above the state of water at the feed temperature t_0, instead of above 32 deg. See § 13 (*d*).

q Table II, column 6. Heat of the liquid, or heat required to raise one pound of water from 32 deg. to the vaporization temperature t; § 13 (*b*).

R Constant or coefficient in the fundamental equation $pv = RT$; for steam (if a perfect gas) the value is 0.5956; see § 12 (*e*).

R/p Table II, column 4. Ideal rate of expansion of superheated steam, or increment of volume per degree under the constant pressure p: to be used in calculating steam volume, as explained in § 12 (*i*).

r Table II, column 7. Latent heat or heat of vaporization, required for the complete vaporization of one pound of water at pressure p and temperature t; § 13 (*c*).

s Table II, column 2. The specific volume of saturated steam (dry or completely vaporized), or the volume of one pound of steam in cubic feet; § 12 (*c*).

s Later, in Chapter VI, § 27, etc., s is used as a symbol for degrees of superheat or for $(t - t_s)$ — this where there is no danger of confusing it with specific volume.

s' Table II, column 3. The ideal specific volume of saturated steam, if it were a perfect gas under the law $pv = RT$: used in calculating the superheated volume v, as explained in § 12 (*g*) and (*i*).

T Absolute temperature, $t + 460$ or, more exactly, $t + 459.6$; § 12 (*g*).

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t Table I, Table II, column 0, etc. Temperature of saturated steam, or of steam formation under the pressure p. Also used

t_s for temperature in general, as of superheated steam, with t_s for the particular value at saturation; § 13 (*h*).

u Volume increase during vaporization, under constant pressure p, from water volume w to steam volume s, in cubic feet per pound of steam; § 12 (*d*).

v Specific volume in general, of steam wet or superheated; saturation volume s is a particular value of v.

v Table VI. Specific volume of superheated steam; see § 12 (*i*) for method of calculation, etc.

v' Ideal specific volume of superheated steam, under the law $pv = RT$; used in getting the actual volume v, as explained in § 12 (*i*).

w Table III, column 2. Specific water volume, or volume in cubic feet of one pound of water at temperature t and pressure p; § 12 (*c*).

x Proportion of steam in a mixture of water and steam, or the weight of steam in one pound of the mixture; the "quality," "quality fraction," or "degree of dryness."

C. Accuracy of the Steam Tables

In regard to this mass of quantitative information, two questions arise. First, are the values essentially correct, or do they represent physical fact within the attainable degree of accuracy in determination? Second, are the successive numbers accurately spaced, so as to represent smoothly and precisely the laws of variation of the quantities? For the second requirement, far greater numerical exactness is needed than for the first.

The simplest property of saturated steam is the fundamental pressure-temperature relation, and this has been determined with a high degree of accuracy, even up to the critical temperature. Of the quantities used in thermodynamic calculations, the most important are specific volume and total heat. In these the range of experimental irregularity, or the departure of individual determinations from the mean curve, varies from 1 in 1000 to 1 in 400, on either side: this statement applies up to about 200 lb. pressure, while above that limit there are few direct data, but an increasing degree of extrapolation, inference, and uncertainty.

The best that the combination and interpretation of physical data can do, in fixing a law of relation between total heat and pressure or

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TABLE I. TEMPERATURES FOR VARIOUS PRESSURES.

Argument, pressure in pounds per square inch above zero.
Body of table, steam temperature in degrees fahrenheit.

p	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0		35.03	53.15	64.50	72.91	79.66	85.32	90.19	94.49	98.34
1	101.84	105.06	108.02	110.78	113.36	115.78	118.07	120.24	122.30	124.27
2	126.15	127.95	129.68	131.34	132.95	134.50	136.00	137.44	138.84	140.20
3	141.52	142.80	144.05	145.27	146.46	147.61	148.74	149.84	150.92	151.97
4	153.00	154.01	155.00	155.97	156.92	157.85	158.77	159.67	160.55	161.42
5	162.27	163.11	163.93	164.74	165.54	166.32	167.09	167.85	168.60	169.34
6	170.07	170.79	171.50	172.20	172.89	173.57	174.24	174.90	175.56	176.21
7	176.85	177.48	178.10	178.72	179.33	179.93	180.53	181.12	181.71	182.29
8	182.86	183.42	183.98	184.54	185.09	185.63	186.17	186.70	187.23	187.75
9	188.27	188.78	189.29	189.80	190.30	190.79	191.28	191.77	192.25	192.73
	0	1	2	3	4	5	6	7	8	9
10	193.21	197.75	201.95	205.87	209.56	213.03	216.32	219.43	222.40	225.24
20	227.96	230.57	233.08	235.49	237.82	240.07	242.25	244.36	246.41	248.40
30	250.34	252.22	254.06	255.84	257.58	259.29	260.96	262.58	264.17	265.73
40	267.26	268.75	270.22	271.66	273.07	274.46	275.82	277.16	278.47	279.76
50	281.03	282.28	283.51	284.72	285.91	287.09	288.25	289.39	290.52	291.63
60	292.73	293.81	294.88	295.93	296.97	298.00	299.02	300.02	301.01	301.99
70	302.95	303.91	304.86	305.79	306.71	307.63	308.54	309.43	310.32	311.20
80	312.07	312.93	313.78	314.62	315.46	316.29	317.11	317.92	318.72	319.52
90	320.31	321.10	321.88	322.65	323.41	324.17	324.92	325.66	326.40	327.13
100	327.86	328.58	329.30	330.01	330.71	331.41	332.11	332.80	333.48	334.16
110	334.83	335.50	336.16	336.82	337.47	338.12	338.77	339.41	340.05	340.68
120	341.31	341.94	342.56	343.18	343.79	344.40	345.00	345.60	346.20	346.79
130	347.38	347.97	348.55	349.13	349.71	350.28	350.85	351.42	351.98	352.54
140	353.10	353.65	354.20	354.75	355.29	355.83	356.37	356.91	357.44	357.97
150	358.49	359.02	359.54	360.06	360.58	361.09	361.60	362.11	362.62	363.12
160	363.62	364.12	364.61	365.11	365.60	366.09	366.57	367.06	367.54	368.02
170	368.50	368.97	369.45	369.92	370.39	370.85	371.32	371.78	372.24	372.70
180	373.15	373.61	374.06	374.51	374.96	375.41	375.85	376.30	376.74	377.18
190	377.62	378.05	378.49	378.92	379.35	379.78	380.20	380.63	381.05	381.47
200	381.89	382.31	382.73	383.14	383.56	383.97	384.38	384.79	385.20	385.60
210	386.01	386.41	386.81	387.21	387.61	388.01	388.40	388.80	389.19	389.58
	0	2	4	6	8	0	2	4	6	8
220	389.97	390.75	391.52	392.29	393.05	393.80	394.55	395.30	396.04	396.78
240	397.51	398.23	398.95	399.67	400.38	401.09	401.79	402.49	403.18	403.87
260	404.56	405.24	405.92	406.59	407.26	407.93	408.59	409.25	409.90	410.55
280	411.20	411.84	412.49	413.13	413.76	414.39	415.01	415.64	416.26	416.87
300	417.49	418.10	418.71	419.31	419.91	420.51	421.11	421.70	422.29	422.87
320	423.46	424.04	424.62	425.19	425.76	426.33	426.90	427.46	428.03	428.59
340	429.14	429.70	430.25	430.80	431.35	431.89	432.43	432.97	433.51	434.05
360	434.58	435.11	435.64	436.17	436.69	437.21	437.73	438.25	438.77	439.28
380	439.79	440.30	440.81	441.31	441.82	442.32	442.82	443.32	443.81	444.31

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TABLE I. TEMPERATURES FOR VARIOUS PRESSURES — *Concluded.*

p	0 50	5 55	10 60	15 65	20 70	25 75	30 80	35 85	40 90	45 95
400	444.80	446.02	447.22	448.42	449.61	450.79	451.95	453.10	454.25	455.38
450	456.51	457.63	458.73	459.83	460.92	462.00	463.07	464.13	465.18	466.23
500	467.27	468.30	469.32	470.33	471.33	472.33	473.32	474.31	475.29	476.26
550	477.22	478.18	479.13	480.07	481.01	481.94	482.86	483.78	484.69	485.60
	0	10	20	30	40	50	60	70	80	90
600	486.50	488.28	490.04	491.78	493.50	495.19	496.87	498.53	500.17	501.79
700	503.39	504.97	506.53	508.08	509.61	511.13	512.63	514.11	515.58	517.04
800	518.48	519.91	521.31	522.72	524.11	525.48	526.84	528.19	529.52	530.85
900	532.16	533.46	534.75	536.03	537.29	538.55	539.80	541.04	542.26	543.48
1000	544.69									

temperature for instance, is to locate the curve within a band from 1 to 2 B.t.u. wide up to 200 lb. pressure, which runs to perhaps 5 or 10 B.t.u. in width at 1000 lb. pressure. Closer delineation is a problem in graphical and mathematical method — the most useful scheme being to choose a simple mathematical formula which agrees fairly well with the trend of experiment, then plot to a large scale the differences between formula and observation, and trace a curve on this plane.

As a statement of physical fact, a value of total heat carried to one decimal place is of the fullest degree of precision that our knowledge will justify, running one significant figure farther than what is surely known as to the absolute, local value of the quantity. In all calculations where observations upon actual steam apparatus enter as data, the last figure or decimal place should be dropped from the numbers in Table I and in columns 1 and 6 to 13 of Table II. There are, however, certain calculations, as of the form of the ideal steam jet, where the result sought depends upon the relatively small difference between large total quantities, and it is to meet this case that the numbers in the tables are carried to five or six significant figures.

In reading values for superheated steam from the diagrams in Tables VII and VIII, an error of 0.5 B.t.u. or its equivalent can easily be made. In comparison with a closeness of 0.1 B.t.u. for saturated steam, this about represents the difference in accuracy of physical data. With the original, full-size diagrams, an accuracy of 0.1 to 0.2 B.t.u. was attainable; but there is an uncertainty in the relative spacing of the curves which is of at least this order of magnitude in the best part of the table, and which reaches perhaps one per cent (of the total heat) at the upper pressure limit of the diagrams. While close enough for all "practical" purposes, these diagrams do not give very satisfactory service in the calculations which require precisely expressed data.

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TABLE II. PRINCIPAL STEAM TABLE

0	1	2	3	4	5	6	7
Temp. deg. fahr. t	Press. lb. per sq. in. p	Sp. vol. cu. ft. per lb. s	Sp. vol. ideal. s'	See definitions. R/p	f_1	Heat of liquid B.t.u. q	Latent heat B.t.u. r
32	0.0886	3304	3305	6.723	1.310	0.00	1072.75
33	0.0922	3180	3181	6.459	1.301	1.01	1072.21
34	0.0960	3062	3063	6.206	1.292	2.01	1071.67
35	0.0999	2948	2949	5.963	1.283	3.02	1071.13
36	0.1039	2839	2840	5.731	1.275	4.02	1070.60
37	0.1081	2734	2735	5.508	1.266	5.03	1070.06
38	0.1125	2633	2634	5.294	1.258	6.03	1069.52
39	0.1170	2537	2538	5.090	1.249	7.04	1068.98
40	0.1217	2444	2445	4.895	1.241	8.04	1068.44
41	0.1265	2356	2357	4.708	1.233	9.04	1067.90
42	0.1315	2271	2272	4.529	1.224	10.05	1067.36
43	0.1367	2189	2190	4.358	1.216	11.05	1066.82
44	0.1420	2111	2112	4.194	1.208	12.05	1066.28
45	0.1475	2036	2037	4.038	1.200	13.05	1065.74
46	0.1532	1964	1965	3.888	1.192	14.05	1065.20
47	0.1591	1895	1896	3.744	1.185	15.05	1064.66
48	0.1652	1829	1830	3.605	1.177	16.05	1064.12
49	0.1715	1765	1766	3.473	1.169	17.05	1063.57
50	0.1780	1704	1705	3.346	1.162	18.05	1063.03
51	0.1847	1645	1646	3.224	1.154	19.05	1062.49
52	0.1917	1588	1589	3.107	1.147	20.05	1061.95
53	0.1989	1533	1534	2.994	1.139	21.05	1061.41
54	0.2063	1481	1482	2.886	1.132	22.05	1060.86
55	0.2140	1431	1432	2.783	1.124	23.05	1060.32
56	0.2219	1383	1384	2.684	1.117	24.05	1059.78
57	0.2301	1337	1338	2.589	1.110	25.05	1059.24
58	0.2385	1292	1293	2.498	1.102	26.05	1058.69
59	0.2472	1249	1250	2.410	1.095	27.04	1058.15
60	0.2561	1207	1208	2.326	1.088	28.04	1057.61
61	0.2653	1167	1168	2.245	1.081	29.04	1057.06
62	0.2749	1129	1130	2.167	1.074	30.04	1056.52
63	0.2847	1092	1093	2.092	1.067	31.03	1055.98
64	0.2948	1057	1058	2.020	1.060	32.03	1055.43
65	0.3053	1024	1025	1.951	1.053	33.03	1054.89
66	0.3161	990.9	992.0	1.8843	1.046	34.03	1054.34
67	0.3272	958.7	959.8	1.8205	1.040	35.02	1053.80
68	0.3386	927.7	928.7	1.7591	1.033	36.02	1053.26
69	0.3504	897.9	898.9	1.7000	1.026	37.02	1052.71
70	0.3625	869.2	870.2	1.6431	1.020	38.01	1052.17
71	0.3750	841.6	842.5	1.5883	1.013	39.01	1051.63
72	0.3879	815.0	815.9	1.5355	1.007	40.01	1051.08
73	0.4012	789.5	790.4	1.4847	1.000	41.00	1050.54
74	0.4148	765.0	765.9	1.4358	0.994	42.00	1049.99
75	0.4289	741.4	742.4	1.3887	0.988	43.00	1049.44

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PRINCIPAL STEAM TABLE TABLE II.

8	9	10	11	12	13	0
Total heat B.t.u.	Inner latent heat.	Inner total heat.	Entropy of Liquid.	Vapor'n.	Entropy total.	Temp. deg. fahr.
H	l	K	a	b	N	t
1072.75	1018.58	1018.58	0.00000	2.18216	2.18216	32
1073.22	1017.93	1018.94	0.00205	2.17663	2.17868	33
1073.68	1017.28	1019.29	0.00409	2.17113	2.17522	34
1074.15	1016.63	1019.65	0.00613	2.16565	2.17178	35
1074.62	1015.99	1020.01	0.00816	2.16020	2.16836	36
1075.09	1015.34	1020.37	0.01018	2.15477	2.16495	37
1075.55	1014.69	1020.72	0.01220	2.14936	2.16156	38
1076.02	1014.04	1021.08	0.01421	2.14397	2.15818	39
1076.48	1013.39	1021.43	0.01622	2.13859	2.15481	40
1076.94	1012.74	1021.78	0.01822	2.13324	2.15146	41
1077.41	1012.09	1022.14	0.02022	2.12791	2.14813	42
1077.87	1011.44	1022.49	0.02222	2.12260	2.14482	43
1078.33	1010.79	1022.84	0.02421	2.11731	2.14152	44
1078.79	1010.14	1023.19	0.02620	2.11204	2.13824	45
1079.25	1009.49	1023.54	0.02818	2.10680	2.13498	46
1079.71	1008.84	1023.89	0.03016	2.10157	2.13173	47
1080.17	1008.19	1024.24	0.03213	2.09637	2.12850	48
1080.62	1007.54	1024.59	0.03410	2.09118	2.12528	49
1081.08	1006.89	1024.94	0.03607	2.08601	2.12208	50
1081.54	1006.24	1025.29	0.03803	2.08087	2.11890	51
1082.00	1005.59	1025.64	0.03999	2.07574	2.11573	52
1082.46	1004.94	1025.99	0.04194	2.07064	2.11258	53
1082.91	1004.28	1026.33	0.04389	2.06555	2.10944	54
1083.37	1003.63	1026.68	0.04583	2.06048	2.10631	55
1083.83	1002.98	1027.03	0.04777	2.05543	2.10320	56
1084.29	1002.33	1027.38	0.04970	2.05040	2.10010	57
1084.74	1001.68	1027.73	0.05163	2.04539	2.09702	58
1085.19	1001.03	1028.07	0.05356	2.04040	2.09396	59
1085.65	1000.38	1028.42	0.05548	2.03543	2.09091	60
1086.10	999.72	1028.76	0.05740	2.03048	2.08788	61
1086.56	999.07	1029.11	0.05931	2.02555	2.08486	62
1087.01	998.42	1029.45	0.06122	2.02063	2.08185	63
1087.46	997.77	1029.80	0.06313	2.01573	2.07886	64
1087.92	997.12	1030.15	0.06503	2.01085	2.07588	65
1088.37	996.46	1030.49	0.06693	2.00599	2.07292	66
1088.82	995.81	1030.83	0.06883	2.00115	2.06998	67
1089.28	995.16	1031.18	0.07072	1.99632	2.06704	68
1089.73	994.51	1031.52	0.07261	1.99151	2.06412	69
1090.18	993.86	1031.87	0.07449	1.98672	2.06121	70
1090.64	993.21	1032.22	0.07637	1.98195	2.05832	71
1091.09	992.55	1032.56	0.07824	1.97720	2.05544	72
1091.54	991.90	1032.90	0.08012	1.97246	2.05258	73
1091.99	991.24	1033.24	0.08199	1.96774	2.04973	74
1092.44	990.58	1033.58	0.08386	1.96304	2.04690	75

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TABLE II. PRINCIPAL STEAM TABLE

0	1	2	3	4	5	6	7
t	p	s	s'	R/p	f_t	q	r
75	0.4289	741.4	742.4	1.3887	0.988	43.00	1049.44
76	0.4433	718.6	719.6	1.3435	0.981	43.99	1048.90
77	0.4582	696.6	697.6	1.2999	0.975	44.99	1048.35
78	0.4735	675.3	676.3	1.2579	0.969	45.99	1047.80
79	0.4893	654.7	655.7	1.2173	0.963	46.98	1047.25
80	0.5055	634.8	635.8	1.1782	0.957	47.98	1046.71
81	0.5222	615.6	616.6	1.1405	0.951	48.97	1046.16
82	0.5394	597.0	598.0	1.1042	0.945	49.97	1045.61
83	0.5570	579.1	580.0	1.0693	0.939	50.97	1045.06
84	0.5752	561.8	562.7	1.0355	0.933	51.96	1044.52
85	0.5939	545.2	546.1	1.0029	0.927	52.96	1043.97
86	0.6132	529.1	530.0	0.9714	0.921	53.95	1043.42
87	0.6330	513.5	514.4	0.9410	0.916	54.95	1042.87
88	0.6533	498.4	499.3	0.9116	0.910	55.95	1042.32
89	0.6743	483.8	484.7	0.8833	0.904	56.94	1041.77
90	0.6958	469.6	470.5	0.8560	0.899	57.94	1041.22
91	0.7179	455.9	456.8	0.8296	0.893	58.94	1040.67
92	0.7406	442.7	443.6	0.8041	0.887	59.93	1040.12
93	0.7640	429.9	430.8	0.7795	0.882	60.93	1039.57
94	0.7880	417.5	418.4	0.7558	0.876	61.92	1039.02
95	0.8127	405.5	406.4	0.7329	0.871	62.92	1038.46
96	0.8380	393.9	394.8	0.7108	0.866	63.92	1037.91
97	0.8640	382.7	383.6	0.6894	0.860	64.91	1037.36
98	0.8907	371.9	372.8	0.6687	0.855	65.91	1036.80
99	0.9181	361.5	362.4	0.6487	0.850	66.91	1036.24
100	0.9462	351.4	352.2	0.6294	0.845	67.90	1035.69
101	0.9751	341.6	342.4	0.6107	0.839	68.90	1035.13
102	1.0047	332.1	332.9	0.5927	0.834	69.90	1034.57
103	1.0350	322.9	323.7	0.5753	0.829	70.89	1034.02
104	1.0662	314.0	314.8	0.5585	0.824	71.89	1033.47
105	1.0982	305.4	306.2	0.5423	0.819	72.89	1032.91
106	1.1310	297.0	297.8	0.5266	0.814	73.88	1032.36
107	1.1647	288.9	289.7	0.5114	0.809	74.88	1031.80
108	1.1992	281.1	281.9	0.4967	0.804	75.88	1031.24
109	1.2347	273.5	274.3	0.4824	0.799	76.87	1030.68
110	1.2711	266.1	266.9	0.4686	0.794	77.87	1030.12
111	1.3084	258.9	259.7	0.4552	0.789	78.87	1029.56
112	1.3466	252.0	252.8	0.4422	0.784	79.86	1029.00
113	1.3858	245.3	246.1	0.4297	0.779	80.86	1028.44
114	1.4260	238.8	239.6	0.4176	0.775	81.86	1027.88
115	1.4671	232.5	233.3	0.4059	0.770	82.86	1027.32
116	1.5093	226.4	227.2	0.3946	0.765	83.85	1026.76
117	1.5525	220.5	221.3	0.3836	0.761	84.85	1026.20
118	1.5968	214.7	215.5	0.3730	0.756	85.85	1025.63
119	1.6421	209.1	209.9	0.3627	0.751	86.84	1025.07
120	1.6886	203.7	204.5	0.3527	0.747	87.84	1024.50
121	1.7362	198.4	199.2	0.3430	0.742	88.84	1023.94
122	1.7849	193.3	194.0	0.3336	0.737	89.84	1023.37
123	1.8348	188.3	189.0	0.3246	0.733	90.83	1022.81
124	1.8859	183.5	184.2	0.3158	0.728	91.83	1022.24
125	1.9382	178.8	179.5	0.3073	0.724	92.83	1021.67

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PRINCIPAL STEAM TABLE TABLE II.

8	9	10	11	12	13	0
H	l	K	a	b	N	t
1092.44	990.58	1033.58	0.08386	1.96304	2.04690	75
1092.89	989.93	1033.93	0.08572	1.95836	2.04408	76
1093.34	989.28	1034.27	0.08758	1.95369	2.04127	77
1093.79	988.62	1034.71	0.08943	1.94904	2.03847	78
1094.23	987.96	1034.94	0.09128	1.94440	2.03568	79
1094.68	987.31	1035.28	0.09313	1.93978	2.03291	80
1095.13	986.66	1035.63	0.09497	1.93518	2.03015	81
1095.58	986.00	1035.97	0.09681	1.93060	2.02741	82
1096.03	985.34	1036.31	0.09865	1.92603	2.02468	83
1096.48	984.69	1036.65	0.10048	1.92148	2.02196	84
1096.93	984.04	1036.99	0.10231	1.91694	2.01925	85
1097.37	983.38	1037.33	0.10414	1.91242	2.01656	86
1097.82	982.72	1037.67	0.10596	1.90792	2.01388	87
1098.27	982.06	1038.01	0.10778	1.90343	2.01121	88
1098.71	981.41	1038.35	0.10960	1.89896	2.00856	89
1099.16	980.75	1038.69	0.11142	1.89450	2.00592	90
1099.61	980.09	1039.03	0.11323	1.89006	2.00329	91
1100.05	979.43	1039.37	0.11504	1.88563	2.00067	92
1100.50	978.77	1039.71	0.11684	1.88122	1.99806	93
1100.94	978.11	1040.04	0.11864	1.87683	1.99547	94
1101.38	977.45	1040.37	0.12044	1.87245	1.99289	95
1101.83	976.89	1040.71	0.12224	1.86809	1.99033	96
1102.27	976.13	1041.05	0.12403	1.86374	1.98777	97
1102.71	975.47	1041.38	0.12582	1.85940	1.98522	98
1103.15	974.81	1041.71	0.12761	1.85508	1.98269	99
1103.59	974.15	1042.05	0.12939	1.85077	1.98016	100
1104.03	973.48	1042.38	0.13177	1.84648	1.97765	101
1104.47	972.82	1042.71	0.13295	1.84220	1.97515	102
1104.91	972.16	1043.05	0.13472	1.83794	1.97266	103
1105.36	971.50	1043.39	0.13649	1.83369	1.97018	104
1105.80	970.84	1043.72	0.13826	1.82946	1.96772	105
1106.24	970.18	1044.06	0.14002	1.82524	1.96526	106
1106.68	969.52	1044.39	0.14178	1.82104	1.96282	107
1107.12	968.85	1044.72	0.14354	1.81685	1.96039	108
1107.55	968.18	1045.05	0.14529	1.81267	1.95796	109
1107.99	967.52	1045.39	0.14704	1.80850	1.95554	110
1108.43	966.85	1045.72	0.14879	1.80435	1.95314	111
1108.86	966.18	1046.04	0.15053	1.80021	1.95074	112
1109.30	965.52	1046.38	0.15227	1.79609	1.94836	113
1109.74	964.85	1046.71	0.15401	1.79198	1.94599	114
1110.18	964.19	1047.04	0.15575	1.78789	1.94364	115
1110.61	963.53	1047.37	0.15748	1.78381	1.94129	116
1111.05	962.86	1047.70	0.15921	1.77974	1.93895	117
1111.48	962.19	1048.03	0.16094	1.77568	1.93662	118
1111.91	961.52	1048.36	0.16267	1.77164	1.93431	119
1112.34	960.85	1048.68	0.16439	1.76761	1.93200	120
1112.78	960.18	1049.01	0.16611	1.76359	1.92970	121
1113.21	959.51	1049.34	0.16783	1.75959	1.92742	122
1113.64	958.94	1049.66	0.16954	1.75560	1.92514	123
1114.07	958.17	1049.99	0.17125	1.75162	1.92287	124
1114.50	957.49	1050.31	0.17296	1.74765	1.92061	125

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TABLE II. PRINCIPAL STEAM TABLE

0	1	2	3	4	5	6	7
t	p	s	s'	R/p	f_1	q	r
125	1.938	178.82	179.55	0.3073	0.724	92.83	1021.67
126	1.992	174.33	175.06	0.2990	0.719	93.83	1021.10
127	2.047	169.96	170.68	0.2910	0.715	94.82	1020.54
128	2.103	165.71	166.53	0.2832	0.711	95.82	1019.97
129	2.160	161.58	162.30	0.2757	0.706	96.82	1019.40
130	2.219	157.57	158.28	0.2684	0.702	97.82	1018.83
131	2.279	153.67	154.38	0.2613	0.698	98.82	1018.26
132	2.341	149.88	150.58	0.2544	0.693	99.81	1017.69
133	2.403	146.19	146.89	0.2477	0.689	100.81	1017.12
134	2.468	142.60	143.29	0.2413	0.685	101.81	1016.55
135	2.533	139.12	139.80	0.2351	0.681	102.81	1015.97
136	2.600	135.73	136.41	0.2291	0.676	103.81	1015.40
137	2.669	132.43	133.11	0.2232	0.672	104.81	1014.82
138	2.740	129.23	129.90	0.2175	0.668	105.80	1014.25
139	2.812	126.13	126.80	0.2119	0.664	106.80	1013.68
140	2.885	123.12	123.78	0.20644	0.660	107.80	1013.10
141	2.960	120.19	120.85	0.20122	0.656	108.80	1012.52
142	3.037	117.34	118.00	0.19614	0.652	109.80	1011.95
143	3.116	114.56	115.22	0.19120	0.648	110.80	1011.37
144	3.196	111.86	112.51	0.18640	0.644	111.80	1010.79
145	3.278	109.23	109.87	0.18174	0.640	112.80	1010.21
146	3.361	106.67	107.31	0.17621	0.636	113.80	1009.63
147	3.447	104.18	104.82	0.17281	0.632	114.80	1009.05
148	3.534	101.76	102.40	0.16853	0.628	115.79	1008.47
149	3.623	99.41	100.04	0.16437	0.624	116.79	1007.89
150	3.715	97.13	97.76	0.16033	0.620	117.79	1007.31
151	3.808	94.91	95.53	0.15641	0.616	118.79	1006.73
152	3.903	92.74	93.36	0.15260	0.613	119.79	1006.15
153	4.000	90.63	91.21	0.14890	0.609	120.79	1005.56
154	4.099	88.57	89.18	0.14531	0.605	121.79	1004.98
155	4.200	86.57	87.18	0.14182	0.601	122.79	1004.39
156	4.303	84.62	85.22	0.13842	0.598	123.79	1003.80
157	4.408	82.72	83.32	0.13511	0.594	124.79	1003.22
158	4.516	80.87	81.46	0.13189	0.590	125.79	1002.63
159	4.625	79.07	79.66	0.12876	0.587	126.79	1002.04
160	4.737	77.32	77.90	0.12572	0.583	127.79	1001.45
161	4.852	75.61	76.19	0.12276	0.580	128.80	1000.86
162	4.968	73.94	74.52	0.11988	0.576	129.80	1000.27
163	5.087	72.32	72.89	0.11708	0.573	130.80	999.68
164	5.208	70.74	71.31	0.11435	0.569	131.80	999.09
165	5.332	69.20	69.77	0.11170	0.566	132.80	998.49
166	5.459	67.70	68.27	0.10911	0.563	133.80	997.90
167	5.588	66.24	66.80	0.10659	0.559	134.80	997.31
168	5.719	64.81	65.37	0.10414	0.556	135.80	996.71
169	5.853	63.42	63.98	0.10175	0.553	136.80	996.12
170	5.990	62.06	62.62	0.09943	0.549	137.80	995.52
171	6.129	60.73	61.29	0.09717	0.546	138.81	994.92
172	6.271	59.44	59.99	0.09497	0.543	139.81	994.33
173	6.416	58.18	58.73	0.09283	0.540	140.81	993.73
174	6.564	56.95	57.49	0.09074	0.536	141.81	993.13
175	6.714	55.75	56.29	0.08871	0.533	142.82	992.53

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PRINCIPAL STEAM TABLE TABLE II.

8	9	10	11	12	13	0
H	l	K	a	b	N	t
1114.50	957.49	1050.31	0.17296	1.74765	1.92061	125
1114.93	956.82	1050.64	0.17467	1.74370	1.91837	126
1115.36	956.15	1050.96	0.17637	1.73976	1.91613	127
1115.79	955.48	1051.29	0.17807	1.73583	1.91390	128
1116.22	954.80	1051.61	0.17977	1.73191	1.91168	129
1116.65	954.13	1051.94	0.18146	1.72801	1.90947	130
1117.08	953.45	1052.26	0.18315	1.72412	1.90727	131
1117.50	952.78	1052.58	0.18484	1.72024	1.90508	132
1117.93	952.10	1052.90	0.18653	1.71637	1.90290	133
1118.36	951.43	1053.23	0.18821	1.71251	1.90072	134
1118.78	950.75	1053.55	0.18989	1.70867	1.89856	135
1119.21	950.07	1053.87	0.19157	1.70484	1.89641	136
1119.63	949.39	1054.19	0.19324	1.70102	1.89426	137
1120.05	948.71	1054.51	0.19491	1.69721	1.89212	138
1120.48	948.04	1054.83	0.19658	1.69341	1.88999	139
1120.90	947.36	1055.15	0.19825	1.68963	1.88788	140
1121.32	946.68	1055.47	0.19991	1.68586	1.88577	141
1121.75	946.00	1055.79	0.20157	1.68210	1.88367	142
1122.17	945.32	1056.11	0.20323	1.67835	1.88158	143
1122.59	944.64	1056.43	0.20489	1.67461	1.87950	144
1123.01	943.96	1056.75	0.20654	1.67088	1.87742	145
1123.43	943.27	1057.06	0.20819	1.66716	1.87535	146
1123.85	942.59	1057.38	0.20984	1.66345	1.87329	147
1124.26	941.91	1057.69	0.21149	1.65976	1.87125	148
1124.68	941.23	1058.01	0.21314	1.65608	1.86922	149
1125.10	940.55	1058.33	0.21478	1.65241	1.86719	150
1125.52	939.86	1058.64	0.21642	1.64875	1.86517	151
1125.94	939.18	1058.96	0.21806	1.64510	1.86316	152
1126.35	938.49	1059.27	0.21969	1.64146	1.86115	153
1126.77	937.81	1059.59	0.22132	1.63783	1.85915	154
1127.18	937.12	1059.90	0.22295	1.63421	1.85716	155
1127.59	936.43	1060.21	0.22458	1.63060	1.85518	156
1128.01	935.74	1060.52	0.22620	1.62700	1.85320	157
1128.42	935.05	1060.83	0.22782	1.62341	1.85123	158
1128.83	934.36	1061.14	0.22944	1.61984	1.84928	159
1129.24	933.67	1061.45	0.23105	1.61628	1.84733	160
1129.66	932.98	1061.77	0.23266	1.61273	1.84539	161
1130.07	932.30	1062.08	0.23427	1.60918	1.84345	162
1130.48	931.61	1062.39	0.23588	1.60565	1.84153	163
1130.89	930.92	1062.70	0.23749	1.60212	1.83961	164
1131.29	930.22	1063.00	0.23909	1.59861	1.83770	165
1131.70	929.53	1063.31	0.24069	1.59511	1.83580	166
1132.11	928.84	1063.62	0.24229	1.59161	1.83390	167
1132.51	928.14	1063.92	0.24389	1.58813	1.83202	168
1132.92	927.45	1064.23	0.24549	1.58465	1.83014	169
1133.32	926.75	1064.53	0.24708	1.58119	1.82827	170
1133.73	926.05	1064.84	0.24867	1.57774	1.82641	171
1134.14	925.36	1065.15	0.25026	1.57430	1.82456	172
1134.54	924.66	1065.45	0.25184	1.57087	1.82271	173
1134.94	923.96	1065.75	0.25342	1.56744	1.82086	174
1135.35	923.27	1066.07	0.25500	1.56402	1.81902	175

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TABLE II. PRINCIPAL STEAM TABLE

0	1	2	3	4	5	6	7
t	p	s	s'	R/p	f_1	q	r
175	6.714	55.75	56.29	0.08871	0.533	142.82	992.53
176	6.867	54.58	55.12	0.08673	0.530	143.82	991.93
177	7.024	53.44	53.97	0.08480	0.527	144.82	991.33
178	7.183	52.33	52.86	0.08292	0.523	145.82	990.73
179	7.345	51.25	51.77	0.08109	0.520	146.83	990.12
180	7.511	50.20	50.72	0.07930	0.517	147.83	989.51
181	7.679	49.17	49.69	0.07756	0.514	148.83	988.91
182	7.850	48.17	48.68	0.07587	0.511	149.84	988.30
183	8.025	47.19	47.70	0.07422	0.508	150.84	987.70
184	8.203	46.23	46.74	0.07261	0.504	151.84	987.09
185	8.384	45.29	45.80	0.07104	0.501	152.85	986.48
186	8.568	44.38	44.88	0.06951	0.498	153.85	985.87
187	8.756	43.49	43.99	0.06802	0.495	154.85	985.26
188	8.947	42.62	43.12	0.06657	0.492	155.86	984.65
189	9.142	41.77	42.26	0.06515	0.489	156.86	984.04
190	9.340	40.93	41.42	0.06377	0.486	157.87	983.42
191	9.542	40.12	40.61	0.06242	0.483	158.87	982.81
192	9.747	39.33	39.82	0.06110	0.480	159.88	982.19
193	9.956	38.56	39.04	0.05982	0.477	160.88	981.58
194	10.169	37.80	38.28	0.05857	0.474	161.88	980.97
195	10.385	37.06	37.54	0.05735	0.472	162.89	980.35
196	10.606	36.34	36.82	0.05616	0.469	163.89	979.74
197	10.830	35.64	36.11	0.05500	0.466	164.90	979.12
198	11.058	34.95	35.42	0.05386	0.463	165.90	978.50
199	11.291	34.28	34.75	0.05275	0.460	166.91	977.88
200	11.527	33.62	34.09	0.05167	0.458	167.92	977.25
201	11.767	32.98	33.44	0.05061	0.455	168.92	976.63
202	12.012	32.35	32.81	0.04958	0.452	169.93	976.00
203	12.261	31.74	32.20	0.04857	0.450	170.94	975.37
204	12.514	31.14	31.59	0.04759	0.447	171.94	974.75
205	12.771	30.55	31.00	0.04663	0.444	172.95	974.12
206	13.033	29.97	30.42	0.04569	0.442	173.95	973.50
207	13.299	29.41	29.86	0.04478	0.439	174.96	972.87
208	13.569	28.86	29.30	0.04389	0.437	175.97	972.24
209	13.844	28.32	28.76	0.04302	0.434	176.97	971.61
210	14.124	27.80	28.24	0.04217	0.432	177.98	970.97
211	14.408	27.28	27.72	0.04134	0.429	178.99	970.34
212	14.697	26.78	27.22	0.04052	0.427	180.00	969.70
213	14.991	26.29	26.72	0.03973	0.424	181.01	969.06
214	15.290	25.81	26.24	0.03895	0.422	182.01	968.43
215	15.593	25.34	25.77	0.03819	0.420	183.02	967.79
216	15 901	24.88	25.31	0.03745	0.417	184.03	967.15
217	16.215	24.43	24.85	0.03673	0.415	185.04	966.51
218	16.534	23.99	24.41	0.03602	0.413	186.05	965.86
219	16.858	23.56	23.98	0.03533	0.410	187.06	965.22
220	17.187	23.13	23.55	0.03465	0.408	188.07	964.58
221	17.522	22.72	23.14	0.03399	0.406	189.08	963.93
222	17.862	22.32	22.73	0.03334	0.403	190.09	963.29
223	18.207	21.92	22.33	0.03271	0.401	191.10	962.64
224	18.558	21.53	21.94	0.03209	0.399	192.11	961.99
225	18.915	21.15	21.56	0.03148	0.396	193.12	961.35

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PRINCIPAL STEAM TABLE TABLE II.

8	9	10	11	12	13	0
H	l	K	a	b	N	t
1135.35	923.27	1066.07	0.25500	1.56402	1.81902	175
1135.75	922.57	1066.37	0.25658	1.56061	1.81719	176
1136.15	921.87	1066.67	0.25816	1.55722	1.81538	177
1136.55	921.17	1066.97	0.25973	1.55383	1.81356	178
1136.95	920.46	1067.27	0.26130	1.55045	1.81175	179
1137.34	919.76	1067.57	0.26287	1.54708	1.80995	180
1137.74	919.06	1067.87	0.26444	1.54372	1.80816	181
1138.14	918.35	1068.17	0.26600	1.54037	1.80637	182
1138.54	917.65	1068.47	0.26756	1.53702	1.80458	183
1138.93	916.94	1068.76	0.26912	1.53369	1.80281	184
1139.33	916.24	1069.06	0.27068	1.53037	1.80105	185
1139.72	915.53	1069.36	0.27224	1.52706	1.79930	186
1140.11	914.82	1069.65	0.27379	1.52376	1.79755	187
1140.51	914.12	1069.95	0.27534	1.52046	1.79580	188
1140.90	913.41	1070.24	0.27689	1.51717	1.79406	189
1141.29	912.70	1070.54	0.27844	1.51389	1.79233	190
1141.68	911.99	1070.83	0.27999	1.51062	1.79061	191
1142.07	911.27	1071.12	0.28153	1.50736	1.78889	192
1142.46	910.56	1071.41	0.28307	1.50411	1.78718	193
1142.85	909.86	1071.71	0.28461	1.50086	1.78547	194
1143.24	909.14	1072.00	0.28615	1.49763	1.78378	195
1143.63	908.43	1072.29	0.28768	1.49440	1.78208	196
1144.01	907.72	1072.58	0.28921	1.49118	1.78039	197
1144.40	907.00	1072.87	0.29074	1.48797	1.77871	198
1144.78	906.29	1073.16	0.29227	1.48477	1.77704	199
1145.17	905.57	1073.45	0.29380	1.48157	1.77537	200
1145.55	904.85	1073.74	0.29532	1.47838	1.77370	201
1145.93	904.13	1074.02	0.29684	1.47520	1.77204	202
1146.31	903.41	1074.31	0.29836	1.47203	1.77039	203
1146.69	902.69	1074.59	0.29988	1.46887	1.76875	204
1147.07	901.97	1074.88	0.30140	1.46572	1.76712	205
1147.45	901.25	1075.16	0.30292	1.46258	1.76550	206
1147.83	900.53	1075.45	0.30443	1.45944	1.76387	207
1148.21	899.81	1075.73	0.30594	1.45631	1.76225	208
1148.58	899.08	1076.01	0.30745	1.45319	1.76064	209
1148.95	898.25	1076.29	0.30896	1.45008	1.75904	210
1149.33	897.63	1076.57	0.31047	1.44697	1.75744	211
1149.70	896.90	1076.85	0.31198	1.44387	1.75585	212
1150.07	896.17	1077.13	0.31348	1.44078	1.75426	213
1150.44	895.44	1077.40	0.31498	1.43770	1.75268	214
1150.81	894.71	1077.68	0.31648	1.43462	1.75110	215
1151.18	893.98	1077.96	0.31797	1.43155	1.74952	216
1151.55	893.25	1078.24	0.31946	1.42849	1.74795	217
1151.91	892.51	1078.51	0.32095	1.42544	1.74639	218
1152.28	891.77	1078.78	0.32244	1.42239	1.74483	219
1152.65	891.04	1079.06	0.32392	1.41935	1.74327	220
1153.01	890.30	1079.33	0.32540	1.41632	1.74172	221
1153.38	889.57	1079.60	0.32688	1.41329	1.74017	222
1153.74	888.83	1079.87	0.32836	1.41027	1.73863	223
1154.10	888.09	1080.14	0.32984	1.40726	1.73710	224
1154.47	887.36	1080.42	0.33132	1.40426	1.73558	225

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TABLE II. PRINCIPAL STEAM TABLE

0	1	2	3	4	5	6	7
t	p	s	s'	R/p	f_1	q	r
225	18.92	21.152	21.559	.03148	0.396	193.12	961.35
226	19.28	20.781	21.185	.03088	0.394	194.13	960.70
227	19.64	20.417	20.819	.03031	0.392	195.14	960.05
228	20.01	20.061	20.461	.02975	0.390	196.15	959.40
229	20.39	19.712	20.109	.02920	0.388	197.16	958.75
230	20.78	19.370	19.766	.02867	0.385	198.17	958.09
231	21.17	19.035	19.428	.02814	0.383	199.18	957.44
232	21.57	18.707	19.098	.02762	0.381	200.19	956.78
233	21.97	18.386	18.776	.02711	0.379	201.20	956.13
234	22.38	18.071	18.459	.02661	0.377	202.22	955.47
235	22.80	17.763	18.149	.02612	0.375	203.23	954.81
236	23.22	17.461	17.846	.02565	0.372	204.24	954.15
237	23.65	17.165	17.548	.02519	0.370	205.25	953.48
238	24.08	16.875	17.256	.02474	0.368	206.27	952.82
239	24.52	16.591	16.971	.02430	0.366	207.28	952.16
240	24.97	16.312	16.690	.02386	0.364	208.29	951.49
241	25.42	16.039	16.415	.02343	0.362	209.31	950.82
242	25.88	15.771	16.145	.02301	0.360	210.32	950.16
243	26.35	15.509	15.881	.02260	0.358	211.33	949.49
244	26.83	15.252	15.623	.02220	0.356	212.35	948.82
245	27.31	15.000	15.369	.02181	0.354	213.36	948.14
246	27.80	14.753	15.020	.02143	0.352	214.38	947.47
247	28.30	14.511	14.876	.02105	0.351	215.39	946.80
248	28.80	14.273	14.637	.02068	0.349	216.40	946.12
249	29.31	14.040	14.402	.02032	0.347	217.42	945.45
250	29.82	13.812	14.172	.019971	0.345	218.43	944.77
251	30.35	13.588	13.947	.019625	0.343	219.45	944.09
252	30.88	13.368	13.725	.019286	0.341	220.47	943.41
253	31.42	13.153	13.508	018954	0.340	221.48	942.73
254	31.97	12.942	13.296	.018629	0.338	222.50	942.04
255	32.52	12.735	13.087	.018311	0.336	223.51	941.36
256	33.09	12.532	12.882	.017999	0.334	224.53	940.68
257	33.66	12.333	12.681	.017693	0.332	225.55	939.99
258	34.24	12.137	12.484	.017394	0.330	226.56	939.31
259	34.83	11.945	12.291	.017101	0.329	227.58	938.62
260	35.42	11.756	12.100	.016814	0.3271	228.59	937.93
261	36.03	11.571	11.914	.016532	0.3254	229.61	937.24
262	36.64	11.390	11.731	.016255	0.3237	230.63	936.54
263	37.26	11.212	11.552	.015984	0.3220	231.65	935.85
264	37.89	11.037	11.375	.015718	0.3203	232.67	935.15
265	38.53	10.865	11.202	.015457	0.3186	233.69	934.46
266	39.18	10.697	11.032	.015202	0.3169	234.70	933.76
267	39.83	10.532	10.866	.014952	0.3152	235.72	933.06
268	40.49	10.370	10.702	.014707	0.3136	236.74	932.37
269	41.17	10.211	10.542	.014467	0.3119	237.76	931.67
270	41.85	10.055	10.385	.014232	0.3103	238.78	930.97
271	42.54	9.902	10.230	.014001	0.3087	239.80	930.27
272	43.24	9.751	10.077	.013774	0.3071	240.82	929.56
273	43.95	9.603	9.928	.013551	0.3055	241.84	928.86
274	44.67	9.458	9.781	.013333	0.3039	242.86	928.15
275	45.40	9.315	9.637	.013119	0.3023	243.88	927.44

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Principal Steam Table — Table II.

8	9	10	11	12	13	0
H	l	K	a.	b	N	t
1154.47	887.36	1080.42	0.33132	1.40426	1.73558	225
1154.83	886.62	1080.69	0.33280	1.40127	1.73407	226
1155.19	885.88	1080.96	0.33427	1.39828	1.73255	227
1155.55	885.14	1081.32	0.33574	1.39530	1.73104	228
1155.91	884.40	1081.50	0.33721	1.39232	1.72953	229
1156.26	883.66	1081.76	0.33868	1.38935	1.72803	230
1156.62	882.92	1082.03	0.34015	1.38639	1.72654	231
1156.97	882.17	1082.29	0.34161	1.38344	1.72505	232
1157.33	881.43	1082.56	0.34307	1.38049	1.72356	233
1157.69	880.69	1082.83	0.34453	1.37755	1.72208	234
1158.04	879.94	1083.10	0.34599	1.37462	1.72061	235
1158.39	879.19	1083.36	0.34745	1.37169	1.71914	236
1158.74	878.44	1083.62	0.34891	1.36877	1.71768	237
1159.09	877.69	1083.88	0.35036	1.36586	1.71622	238
1159.44	876.94	1084.14	0.35181	1.36295	1.71476	239
1159.78	876.19	1084.40	0.35326	1.36005	1.71331	240
1160.13	875.43	1084.66	0.35471	1.35716	1.71187	241
1160.48	874.68	1084.92	0.35616	1.35427	1.71043	242
1160.82	873.93	1085.18	0.35760	1.35139	1.70899	243
1161.17	873.17	1085.44	0.35904	1.34852	1.70756	244
1161.51	872.41	1085.69	0.36048	1.34565	1.70613	245
1161.85	871.66	1085.95	0.36192	1.34279	1.70471	246
1162.19	870.90	1086.20	0.36336	1.33994	1.70330	247
1162.52	870.14	1086.45	0.36480	1.33709	1.70189	248
1162.86	869.38	1086.70	0.36623	1.33425	1.70048	249
1163.20	868.62	1086.96	0.36766	1.33141	1.69907	250
1163.54	867.86	1087.21	0.36909	1.32858	1.69767	251
1163.88	867.10	1087.46	0.37052	1.32576	1.69628	252
1164.21	866.33	1087.71	0.37195	1.32294	1.69489	253
1164.54	865.56	1087.96	0.37337	1.32013	1.69350	254
1164.87	864.80	1088.21	0.37479	1.31733	1.69212	255
1165.21	864.04	1088.46	0.37621	1.31453	1.69074	256
1165.54	863.27	1088.71	0.37763	1.31174	1.68937	257
1165.87	862.51	1088.96	0.37905	1.30895	1.68800	258
1166.20	861.74	1089.21	0.38047	1.30617	1.68664	259
1166.52	860.97	1089.45	0.38188	1.30340	1.68528	260
1166.85	860.20	1089.70	0.38329	1.30063	1.68392	261
1167.17	859.42	1089.94	0.38470	1.29787	1.68257	262
1167.50	858.65	1090.18	0.38611	1.29512	1.68123	263
1167.82	857.87	1090.42	0.38752	1.29237	1.67989	264
1168.14	857.10	1090.66	0.38893	1.28963	1.67856	265
1168.46	856.32	1090.90	0.39033	1.28689	1.67722	266
1168.78	855.55	1091.14	0.39173	1.28416	1.67589	267
1169.11	854.78	1091.39	0.39313	1.28143	1.67456	268
1169.43	854.00	1091.63	0.39453	1.27871	1.67324	269
1169.75	853.22	1091.87	0.39593	1.27600	1.67193	270
1170.07	852.44	1092.11	0.39733	1.27329	1.67062	271
1170.38	851.66	1092.34	0.39873	1.27059	1.66932	272
1170.70	850.88	1092.58	0.40012	1.26789	1.66801	273
1171.01	850.10	1092.82	0.40151	1.26520	1.66671	274
1171.32	849.31	1093.05	0.40290	1.26252	1.66542	275

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TABLE II. PRINCIPAL STEAM TABLE

0	1	2	3	4	5	6	7
t	p	s	s'	R/p	f_1	q	r
275	45.40	9.315	9.637	.013119	0.3023	243.88	927.44
276	46.14	9.175	9.495	.012909	0.3007	244.90	926.73
277	46.89	9.037	9.356	.012703	0.2991	245.92	926.02
278	47.64	8.902	9.219	.012501	0.2976	246.95	925.30
279	48.41	8.770	9.086	.012303	0.2960	247.97	924.59
280	49.19	8.640	8.955	.012108	0.2945	248.99	923.88
281	49.98	8.512	8.826	.011917	0.2930	250.01	923.16
282	50.78	8.386	8.698	.011730	0.2914	251.03	922.45
283	51.58	8.263	8.574	.011546	0.2899	252.06	921.73
284	52.40	8.142	8.451	.011366	0.2884	253.08	921.01
285	53.23	8.023	8.331	.011189	0.2869	254.11	920.29
286	54.07	7.906	8.213	.011015	0.2854	255.13	919.57
287	54.92	7.791	8.096	.010844	0.2839	256.15	918.84
288	55.78	7.678	7.982	.010677	0.2824	257.18	918.12
289	56.65	7.567	7.870	.010513	0.2809	258.20	917.39
290	57.53	7.458	7.760	.010352	0.2794	259.23	916.67
291	58.43	7.351	7.652	.010194	0.2780	260.26	915.94
292	59.33	7.246	7.545	.010039	0.2765	261.28	915.21
293	60.25	7.142	7.540	.009887	0.2751	262.31	914.48
294	61.18	7.040	7.337	.009737	0.2736	263.34	913.75
295	62.11	6.940	7.236	.009590	0.2722	264.36	913.02
296	63.06	6.842	7.137	.009446	0.2707	265.39	912.28
297	64.02	6.745	7.039	.009304	0.2693	266.42	911.55
298	65.00	6.650	6.942	.009164	0.2679	267.45	910.81
299	65.99	6.556	6.847	.009027	0.2664	268.47	910.08
300	66.98	6.464	6.754	.008892	0.2650	269.50	909.34
301	67.99	6.373	6.662	.008760	0.2636	270.53	908.60
302	69.01	6.284	6.572	.008630	0.2622	271.56	907.86
303	70.05	6.197	6.483	.008503	0.2608	272.59	907.11
304	71.09	6.111	6.396	.008378	0.2594	273.61	906.37
305	72.15	6.027	6.311	.008255	0.2580	274.64	905.62
306	73.22	5.944	6.227	.008134	0.2566	275.67	904.88
307	74.31	5.862	6.144	.008015	0.2553	276.70	904.13
308	75.40	5.782	6.063	.007898	0.2539	277.73	903.38
309	76.51	5.703	5.983	.007784	0.2515	278.76	902.63
310	77.64	5.625	5.904	.007672	0.2512	279.79	901.88
311	78.77	5.548	5.826	.007561	0.2498	280.82	901.12
312	79.92	5.473	5.749	.007452	0.2485	281.85	900.36
313	81.08	5.399	5.674	.007345	0.2472	282.89	899.60
314	82.26	5.326	5.600	.007240	0.2458	283.92	898.84
315	83.45	5.255	5.528	.007137	0.2445	284.95	898.08
316	84.65	5.185	5.457	.007036	0.2432	285.98	897.32
317	85.87	5.115	5.387	.006936	0.2419	287.02	896.55
318	87.10	5.047	5.317	.006838	0.2406	288.05	895.79
319	88.34	4.980	5.249	.006742	0.2393	289.08	895.03
320	89.60	4.914	5.182	.006647	0.2380	290.11	894.26
321	90.87	4.849	5.116	.006554	0.2367	291.15	893.49
322	92.16	4.785	5.051	.006463	0.2354	292.18	892.72
323	93.46	4.722	4.987	.006373	0.2341	293.21	891.95
324	94.78	4.660	4.924	.006248	0.2328	294.25	891.17
325	96.11	4.599	4.862	.006197	0.2315	295.28	890.40

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PRINCIPAL STEAM TABLE TABLE II.

8	9	10	11	12	13	0
H	l	K	a	b	N	t
1171.32	849.31	1093.05	0.40290	1.26252	1.66542	275
1171.63	848.53	1093.28	0.40429	1.25984	1.66413	276
1171.94	847.74	1093.51	0.40568	1.25717	1.66285	277
1172.25	846.95	1093.75	0.40707	1.25450	1.66157	278
1172.56	846.16	1093.98	0.40845	1.25183	1.66028	279
1172.87	845.38	1094.21	0.40983	1.24917	1.65900	280
1173.17	844.59	1094.44	0.41121	1.24652	1.65773	281
1173.48	843.80	1094.67	0.41259	1.24387	1.65646	282
1173.79	843.01	1094.90	0.41397	1.24123	1.65520	283
1174.09	842.22	1095.13	0.41535	1.23859	1.65394	284
1174.40	841.42	1095.36	0.41672	1.23596	1.65268	285
1174.70	840.63	1095.59	0.41810	1.23333	1.65143	286
1175.00	839.83	1095.81	0.41947	1.23071	1.65018	287
1175.30	839.03	1096.03	0.42084	1.22810	1.64894	288
1175.60	838.23	1096.26	0.42221	1.22549	1.64770	289
1175.90	837.44	1096.48	0.42358	1.22289	1.64647	290
1176.20	836.64	1096.71	0.42495	1.22029	1.64524	291
1176.49	835.84	1096.93	0.42632	1.21770	1.64402	292
1176.79	835.04	1097.16	0.42768	1.21511	1.64279	293
1177.09	834.24	1097.38	0.42904	1.21253	1.64157	294
1177.38	833.44	1097.60	0.43040	1.20995	1.64035	295
1177.67	832.63	1097.82	0.43176	1.20738	1.63914	296
1177.97	831.83	1098.05	0.43312	1.20481	1.63793	297
1178.26	831.03	1098.27	0.43448	1.20225	1.63673	298
1178.55	830.23	1098.49	0.43584	1.19969	1.63553	299
1178.84	829.42	1098.70	0.43719	1.19713	1.63432	300
1179.13	828.61	1098.92	0.43854	1.19458	1.63312	301
1179.42	827.80	1099.14	0.43989	1.19204	1.63193	302
1179.70	826.99	1099.35	0.44124	1.18950	1.63074	303
1179.98	826.18	1099.66	0.44259	1.18697	1.62956	304
1180.26	825.36	1099.87	0.44394	1.18444	1.62838	305
1180.55	824.55	1099.99	0.44529	1.18192	1.62721	306
1180.83	823.74	1100.20	0.44664	1.17940	1.62604	307
1181.11	822.93	1100.42	0.44798	1.17689	1.62487	308
1181.39	822.12	1100.63	0.44932	1.17438	1.62370	309
1181.67	821.30	1100.84	0.45066	1.17187	1.62253	310
1181.94	820.48	1101.04	0.45200	1.16937	1.62137	311
1182.21	819.66	1101.25	0.45334	1.16688	1.62022	312
1182.49	818.83	1101.46	0.45468	1.16439	1.61907	313
1182.76	818.01	1101.66	0.45602	1.16190	1.61792	314
1183.03	817.19	1101.87	0.45735	1.15942	1.61677	315
1183.30	816.37	1102.07	0.45868	1.15695	1.61563	316
1183.57	815.53	1102.27	0.46001	1.15448	1.61449	317
1183.84	814.71	1102.48	0.46134	1.15201	1.61335	318
1184.11	813.89	1102.68	0.46267	1.14955	1.61222	319
1184.37	813.06	1102.88	0.46400	1.14709	1.61109	320
1184.64	812.23	1103.08	0.46533	1.14464	1.60997	321
1184.90	811.40	1103.28	0.46666	1.14219	1.60885	322
1185.16	810.57	1103.47	0.46798	1.13975	1.60773	323
1185.42	809.74	1103.67	0.46930	1.13731	1.60661	324
1185.68	808.91	1103.87	0.47062	1.13487	1.60549	325

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TABLE II. PRINCIPAL STEAM TABLE

0	1	2	3	4	5	6	7
t	p	s	s'	R/p	f_1	q	r
325	96.11	4.599	4.862	.006197	0.2315	295.28	890.40
326	97.46	4.539	4.801	.006111	0.2302	296.32	889.62
327	98.82	4.480	4.741	.006027	0.2290	297.35	888.85
328	100.19	4.422	4.682	.005944	0 2277	298.39	888.07
329	101.58	4.365	4.624	.005863	0.2264	299.43	887.29
330	102.99	4.308	4.566	.005783	0.2252	300.47	886.51
331	104.41	4.253	4.510	.005704	0.2239	301.50	885.73
332	105.85	4.198	4.454	.005627	0.2227	302.54	884.94
333	107.30	4.144	4.399	.005551	0.2215	303.58	884.15
334	108.77	4.091	4.345	.005476	0.2202	304.62	883.37
335	110.25	4.039	4.293	.005402	0.2190	305.66	882.58
336	111.75	3.988	4.241	.005329	0.2178	306.70	881.79
337	113.27	3.937	4.189	.005258	0.2166	307.74	881.00
338	114.80	3.887	4.138	.005188	0.2154	308.78	880.20
339	116.35	3.838	4.088	.005119	0.2142	309.82	879.41
340	117.92	3.790	4.039	.005051	0.2130	310.86	878.61
341	119.50	3.742	3.990	.004984	0.2118	311.90	877.81
342	121.10	3.695	3.942	.004918	0.2106	312.95	877.01
343	122.72	3.649	3.895	.004853	0.2094	313.99	876.21
344	124.35	3.604	3.849	.004789	0.2083	315.03	875.40
345	126.00	3.559	3.804	.004727	0.2071	316.07	874.60
346	127.67	3.515	3.759	.004665	0.2059	317.11	873.80
347	129.35	3.472	3.715	.004604	0.2048	318.16	872.99
348	131.05	3.429	3.671	.004545	0.2036	319.20	872.18
349	132.77	3.387	3.628	.004486	0.2024	320.24	871.37
350	134.51	3.345	3.585	.004428	0.2013	321.28	870.56
351	136.26	3.304	3.532	.004371	0.2001	322.33	869.74
352	138.03	3.264	3.502	.004315	0.1990	323.37	868.93
353	139.83	3.224	3.461	.004260	0.1979	324.41	868.12
354	141.64	3.185	3.421	.004205	0.1967	325.46	867.30
355	143.47	3.146	3.382	.004151	0.1956	326.50	866.48
356	145.32	3.108	3.343	.004098	0.1945	327.55	865.66
357	147.18	3.071	3.305	.004046	0.1933	328.50	864.83
358	149.06	3.034	3.267	.003995	0.1922	329.64	864.01
359	150.96	2.997	3.230	.003945	0.1911	330.69	863.19
360	152.88	2.962	3.193	.003896	0.1900	331.74	862.36
361	154.82	2.926	3.157	.003847	0.1889	332.79	861.53
362	156.78	2.891	3.121	.003799	0.1878	333.83	860.70
363	158.76	2.857	3.086	.003752	0.1867	334.88	859.87
364	160.76	2.823	3.051	.003707	0.1857	335.93	859.03
365	162.78	2.790	3.017	.003659	0.1846	336.98	858.20
366	164.82	2.757	2.983	.003614	0.1835	338.03	857.36
367	166.88	2.724	2.950	.003570	0.1824	339.08	856.52
368	168.96	2.692	2.917	.003526	0.1814	340.13	855.68
369	171.06	2.661	2.885	.003482	0.1803	341.18	854.84
370	173.18	2.630	2.853	.003439	0.1792	342.23	854.00
371	175.32	2.599	2.822	.003397	0.1782	343.28	853.15
372	177.48	2.569	2.791	.003356	0.1772	344.33	852.31
373	179.66	2.539	2.760	.003315	0.1761	345.39	851.46
374	181.86	2.510	2.730	.003275	0.1751	346.44	850.61
375	184.09	2.481	2.700	.003235	0.1741	347.49	849.76

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PRINCIPAL STEAM TABLE TABLE II.

8	9	10	11	12	13	0
H	*l*	*K*	*a*	*b*	*N*	*t*
1185.68	808.91	1103.87	0.47062	1.13487	1.60549	325
1185.94	808.07	1104.07	0.47194	1.13244	1.60438	326
1186.20	807.24	1104.27	0.47326	1.13001	1.60327	327
1186.46	806.40	1104.46	0.47458	1.12759	1.60217	328
1186.72	805.56	1104.66	0.47589	1.12517	1.60106	329
1186.98	804.73	1104.86	0.47720	1.12275	1.59995	330
1187.23	803.89	1105.05	0.47851	1.12034	1.59885	331
1187.48	803.05	1105.24	0.47982	1.11793	1.59775	332
1187.73	802.21	1105.43	0.48113	1.11553	1.59666	333
1187.99	801.37	1105.62	0.48244	1.11313	1.59557	334
1188.24	800.52	1105.82	0.48375	1.11074	1.59449	335
1188.49	799.68	1106.01	0.48506	1.10835	1.59341	336
1188.74	798.83	1106.20	0.48636	1.10596	1.59232	337
1188.98	797.98	1106.38	0.48767	1.10358	1.59125	338
1189.23	797.13	1106.57	0.48897	1.10120	1.59017	339
1189.47	796.28	1106.75	0.49027	1.09883	1.58910	340
1189.71	795.43	1106.94	0.49157	1.09646	1.58803	341
1189.96	794.58	1107.13	0.49287	1.09409	1.58696	342
1190.20	793.73	1107.31	0.49417	1.09173	1.58590	343
1190.43	792.87	1107.49	0.49547	1.08937	1.58484	344
1190.67	792.02	1107.67	0.49676	1.08702	1.58378	345
1190.91	791.17	1107.85	0.49806	1.08467	1.58273	346
1191.15	790.31	1108.03	0.49935	1.08232	1.58167	347
1191.38	789.45	1108.21	0.50064	1.07998	1.58062	348
1191.61	788.59	1108.39	0.50193	1.07764	1.57957	349
1191.84	787.73	1108.56	0.50322	1.07531	1.57853	350
1192.07	786.87	1108.74	0.50451	1.07298	1.57749	351
1192.30	786.01	1108.92	0.50580	1.07065	1.57645	352
1192.53	785.15	1109.09	0.50708	1.06833	1.57541	353
1192.76	784.28	1109.27	0.50837	1.06601	1.57438	354
1192.98	783.42	1109.44	0.50965	1.06369	1.57334	355
1193.21	782.55	1109.61	0.51093	1.06138	1.57231	356
1193.43	781.67	1109.78	0.51221	1.05907	1.57128	357
1193.65	780.81	1109.95	0.51349	1.05677	1.57026	358
1193.88	779.95	1110.13	0.51477	1.05447	1.56924	359
1194.10	779.07	1110.30	0.51605	1.05217	1.56822	360
1194.32	778.20	1110.47	0.51733	1.04988	1.56721	361
1194.53	777.33	1110.63	0.51860	1.04759	1.56619	362
1194.75	776.46	1110.80	0.51988	1.04530	1.56518	363
1194.96	775.57	1110.96	0.52115	1.04302	1.56417	364
1195.18	774.70	1111.13	0.52242	1.04074	1.56316	365
1195.39	773.82	1111.29	0.52369	1.03847	1.56216	366
1195.60	772.94	1111.46	0.52496	1.03620	1.56116	367
1195.81	772.06	1111.62	0.52623	1.03393	1.56016	368
1196.02	771.18	1111.78	0.52750	1.03167	1.55917	369
1196.23	770.30	1111.94	0.52877	1.02941	1.55818	370
1196.43	769.41	1112.10	0.53004	1.02715	1.55719	371
1196.64	768.53	1112.26	0.53130	1.02490	1.55620	372
1196.85	767.65	1112.43	0.53257	1.02265	1.55522	373
1197.05	766.76	1112.58	0.53383	1.02040	1.55423	374
1197.25	765.87	1112.74	0.53509	1.01816	1.55325	375

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TABLE II. PRINCIPAL STEAM TABLE

0	1	2	3	4	5	6	7
t	p	s	s'	R/p	f_1	q	r
375	184.09	2.481	2.700	.003235	0.1741	347.49	849.76
376	186.33	2.452	2.671	.003196	0.1730	348.55	848.90
377	188.60	2.424	2.642	.003158	0.1720	349.60	848.05
378	190.88	2.396	2.613	.003120	0.1710	350.65	847.19
379	193.19	2.369	2.585	.003083	0.1700	351.70	846.34
380	195.52	2.342	2.558	.003046	0.1689	352.76	845.48
381	197.87	2.315	2.530	.003010	0.1679	353.82	844.62
382	200.25	2.289	2.503	.002974	0.1669	354.87	843.76
383	202.65	2.263	2.476	.002939	0.1659	355.93	842.89
384	205.07	2.237	2.450	.002905	0.1649	356.98	842.03
385	207.51	2.212	2.424	.002870	0.1639	358.04	841.16
386	209.97	2.187	2.398	.002837	0.1629	359.10	840.29
387	212.46	2.162	2.373	.002803	0.1619	360.16	839.42
388	214.97	2.138	2.348	.002771	0.1609	361.22	838.54
389	217.50	2.114	2.324	.002737	0.1600	362.27	837.67
390	220.06	2.090	2.299	.002707	0.1590	363.33	836.79
391	222.64	2.067	2.275	.002675	0.1580	364.39	835.91
392	225.24	2.044	2.252	.002644	0.1571	365.45	835.03
393	227.87	2.021	2.228	.002614	0.1561	366.51	834.15
394	230.52	1.999	2.205	.002584	0.1552	367.57	833.26
395	233.19	1.977	2.183	.002554	0.1543	368.63	832.38
396	235.89	1.955	2.160	.002525	0.1533	369.69	831.49
397	238.61	1.934	2.138	.002496	0.1524	370.76	830.60
398	241.36	1.912	2.116	.002468	0.1515	371.82	829.71
399	244.13	1.892	2.095	.002440	0.1506	372.88	828.81
400	246.93	1.871	2.073	.002412	0.1497	373.94	827.92
401	249.75	1.850	2.052	.002385	0.1488	375.01	827.02
402	252.60	1.830	2.032	.002358	0.1479	376.07	826.12
403	255.47	1.810	2.011	.002331	0.1470	377.13	825.22
404	258.37	1.791	1.991	.002305	0.1461	378.20	824.31
405	261.29	1.771	1.971	.002280	0.1452	379.26	823.41
406	264.24	1.752	1.951	.002254	0.1443	380.33	822.50
407	267.22	1.733	1.932	.002239	0.1434	381.39	821.59
408	270.22	1.715	1.912	.002204	0.1425	382.46	820.68
409	273.25	1.696	1.893	.002180	0.1417	383.53	819.77
410	276.30	1.678	1.875	.002156	0.1408	384.59	818.86
411	279.38	1.660	1.856	.002132	0.1399	385.66	817.94
412	282.48	1.643	1.838	.002108	0.1391	386.73	817.02
413	285.61	1.625	1.820	.002085	0.1382	387.80	816.10
414	288.77	1.608	1.802	.002063	0.1374	388.86	815.18
415	291.96	1.591	1.784	.002040	0.1365	389.93	814.25
416	295.17	1.574	1.767	.002018	0.1357	391.00	813.33
417	298.41	1.557	1.750	.001996	0.1349	392.07	812.40
418	301.68	1.541	1.733	.001974	0.1340	393.14	811.47
419	304.98	1.525	1.716	.001953	0.1332	394.21	810.54
420	308.30	1.509	1.699	.001932	0.1324	395.28	809.61
421	311.65	1.493	1.683	.001911	0.1316	396.36	808.67
422	315.03	1.477	1.667	.001891	0.1308	397.43	807.73
423	318.44	1.462	1.651	.001870	0.1300	398.50	806.79
424	321.88	1.447	1.635	.001850	0.1292	399.57	805.85
425	325.34	1.432	1.620	.001831	0.1284	400.65	804.90

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PRINCIPAL STEAM TABLE TABLE II.

8	9	10	11	12	13	0
H	l	K	a	b	N	t
1197.25	765.87	1112.74	0.53509	1.01816	1.55325	375
1197.45	764.97	1112.89	0.53635	1.01592	1.55227	376
1197.65	764.09	1113.05.	0.53761	1.01368	1.55129	377
1197.84	763.20	1113.20	0.53887	1.01145	1.55032	378
1198.04	762.31	1113.35	0.54013	1.00922	1.54935	379
1198.24	761.41	1113.50	0.54139	1.00700	1.54839	380
1198.44	760.52	1113.66	0.54265	1.00478	1.54743	381
1198.63	759.62	1113.81	0.54390	1.00256	1.54646	382
1198.82	758.72	1113.96	0.54516	1.00034	1.54550	383
1199.01	757.83	1114.11	0.54641	0.99813	1.54454	384
1199.20	756.93	1114.26	0.54766	0.99592	1.54358	385
1199.39	756.03	1114.41	0.54891	0.99371	1.54262	386
1199.58	755.13	1114.56	0.55016	0.99151	1.54167	387
1199.76	754.22	1114.70	0.55141	0.98931	1.54072	388
1199.94	753.32	1114.84	0.55266	0.98711	1.53977	389
1200.12	752.41	1114.98	0.55391	0.98492	1.53883	390
1200.30	751.50	1115.12	0.55516	0.98273	1.53789	391
1200.48	750.59	1115.26	0.55640	0.98054	1.53694	392
1200.66	749.68	1115.40	0.55765	0.97835	1.53600	393
1200.83	748.76	1115.54	0.55889	0.97617	1.53506	394
1201.01	747.85	1115.68	0.56013	0.97399	1.53412	395
1201.18	746.94	1115.82	0.56137	0.97181	1.53318	396
1201.36	746.02	1115.96	0.56261	0.96964	1.53225	397
1201.53	745.10	1116.09	0.56385	0.96747	1.53132	398
1201.69	744.18	1116.22	0.56509	0.96530	1.53039	399
1201.86	743.27	1116.35	0.56633	0.96314	1.52947	400
1202.03	742.35	1116.49	0.56757	0.96098	1.52855	401
1202.19	741.43	1116.62	0.56880	0.95882	1.52762	402
1202.35	740.50	1116.74	0.57004	0.95666	1.52670	403
1202.51	739.57	1116.87	0.57127	0.95451	1.52578	404
1202.67	738.65	1117.00	0.57250	0.95236	1.52486	405
1202.83	737.72	1117.13	0.57373	0.95021	1.52394	406
1202.98	736.79	1117.25	0.57496	0.94807	1.52303	407
1203.14	735.86	1117.38	0.57619	0.94593	1.52212	408
1203.30	734.93	1117.51	0.57742	0.94379	1.52121	409
1203.45	734.01	1117.63	0.57865	0.94165	1.52030	410
1203.60	733.07	1117.75	0.57988	0.93952	1.51940	411
1203.75	732.13	1117.87	0.58111	0.93739	1.51850	412
1203.90	731.19	1117.99	0.58233	0.93526	1.51759	413
1204.04	730.26	1118.11	0.58356	0.93313	1.51669	414
1204.18	729.31	1118.22	0.58478	0.93101	1.51579	415
1204.33	728.38	1118.34	0.58600	0.92889	1.51489	416
1204.47	727.44	1118.46	0.58722	0.92677	1.51399	417
1204.61	726.49	1118.57	0.58844	0.92466	1.51310	418
1204.75	725.55	1118.69	0.58966	0.92255	1.51221	419
1204.89	724.61	1118.81	0.59088	0.92044	1.51132	420
1205.03	723.66	1118.92	0.59210	0.91833	1.51043	421
1205.16	722.71	1119.03	0.59332	0.91632	1.50955	422
1205.29	721.76	1119.14	0.59453	0.91413	1.50866	423
1205.42	720.81	1119.24	0.59575	0.91203	1.50778	424
1205.55	719.85	1119.35	0.59697	0.90993	1.50690	425

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TABLE II. PRINCIPAL STEAM TABLE

0	1	2	3	4	5	6	7
t	p	s	s'	R/p	f_1	q'	r
425	325.3	1.432	1.620	.001831	0.1284	400.65	804.90
426	328.8	1.417	1.604	.001811	0.1276	401.72	803.96
427	332.4	1.402	1.589	.001792	0.1268	402.79	803.01
428	335.9	1.387	1.574	.001773	0.1260	403.87	802.06
429	339.5	1.373	1.559	.001755	0.1252	404.94	801.11
430	343.1	1.359	1.544	.001736	0.1244	406.02	800.16
431	346.7	1.345	1.530	.001718	0.1237	407.10	799.20
432	350.4	1.331	1.516	.001700	0.1229	408.17	798.24
433	354.1	1.317	1.501	.001682	0.1221	409.25	797.28
434	357.8	1.304	1.487	.001665	0.1214	410.33	796.32
435	361.6	1.291	1.474	.001647	0.1206	411.41	795.35
436	365.4	1.278	1.460	.001630	0.1199	412.48	794.39
437	369.2	1.265	1.446	.001613	0.1191	413.56	793.42
438	373.0	1.252	1.433	.001597	0.1184	414.64	792.45
439	376.9	1.239	1.420	.001580	0.1177	415.72	791.48
440	380.8	1.227	1.407	.001564	0.1169	416.80	790.50
441	384.8	1.214	1.394	.001548	0.1162	417.88	789.52
442	388.7	1.202	1.382	.001532	0.1155	418.96	788.54
443	392.7	1.190	1.369	.001517	0.1148	420.04	787.56
444	396.8	1.178	1.357	.001491	0.1140	421.13	786.57
445	400.8	1.166	1.344	.001486	0.1133	422.21	785.59
446	404.9	1.154	1.332	.001471	0.1126	423.29	784.60
447	409.1	1.143	1.320	.001456	0.1119	424.37	783.61
448	413.2	1.131	1.308	.001441	0.1112	425.46	782.61
449	417.4	1.120	1.297	.001427	0.1105	426.54	781.62
450	421.7	1.109	1.285	.001413	0.1098	427.63	780.62
451	425.9	1.098	1.273	.001398	0.1091	428.71	779.62
452	430.2	1.087	1.262	.001384	0.1084	429.80	778.61
453	434.6	1.076	1.251	.001371	0.1077	430.88	777.61
454	438.9	1.066	1.240	.001357	0.1070	431.97	776.60
455	443.3	1.055	1.229	.001344	0.1064	433.05	775.59
456	447.7	1.045	1.218	.001330	0.1057	434.14	774.58
457	452.2	1.034	1.207	.001317	0.1050	435.23	773.56
458	456.7	1.024	1.197	.001304	0.1044	436.32	772.54
459	461.2	1.014	1.186	.001291	0.1037	437.41	771.52
460	465.8	1.004	1.176	.001279	0.1031	438.50	770.50
461	470.4	0.994	1.166	.001266	0.1024	439.59	769.47
462	475.0	0.985	1.156	.001254	0.1018	440.68	768.45
463	479.7	0.975	1.146	.001242	0.1011	441.77	767.42
464	484.4	0.966	1.136	.001230	0.1005	442.86	766.39
465	489.1	0.956	1.126	.001218	0.0999	443.96	765.35
466	493.9	0.947	1.116	.001206	0.0992	445.05	764.31
467	498.7	0.938	1.107	.001194	0.0986	446.14	763.27
468	503.6	0.929	1.097	.001183	0.0980	447.23	762.23
469	508.5	0.920	1.088	.001171	0.0974	448.33	761.18
470	513.4	0.911	1.079	.001160	0.0968	449.42	760.13
471	518.3	0.902	1.069	.001149	0.0962	450.52	759.08
472	523.3	0.893	1.060	.001138	0.0956	451.61	758.03
473	528.4	0.885	1.051	.001127	0.0950	452.71	756.97
474	533.4	0.876	1.042	.001117	0.0944	453.81	755.91
475	538.5	0.868	1.034	.001106	0.0938	454.90	754.85

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PRINCIPAL STEAM TABLE TABLE II.

8	9	10	11	12	13	0
H	l	K	a	b	N	t
1205.55	719.85	1119.35	0.59697	0.90993	1.50690	425
1205.68	718.90	1119.46	0.59818	0.90784	1.50602	426
1205.80	717.95	1119.56	0.59939	0.90575	1.50514	427
1205.93	716.99	1119.67	0.60060	0.90366	1.50426	428
1206.05	716.03	1119.77	0.60181	0.90157	1.50338	429
1206.18	715.08	1119.88	0.60302	0.89948	1.50250	430
1206.30	714.12	1119.98	0.60423	0.89740	1.50163	431
1206.41	713.15	1120.08	0.60544	0.89532	1.50076	432
1206.53	712.19	1120.18	0.60665	0.89324	1.49989	433
1206.65	711.23	1120.29	0.60785	0.89116	1.49901	434
1206.76	710.26	1120.38	0.60906	0.88908	1.49814	435
1206.87	709.30	1120.48	0.61026	0.88701	1.49727	436
1206.98	708.33	1120.57	0.61147	0.88494	1.49641	437
1207.09	707.36	1120.67	0.61267	0.88287	1.49554	438
1207.20	706.39	1120.76	0.61387	0.88080	1.49467	439
1207.30	705.41	1120.85	0.61507	0.87874	1.49381	440
1207.40	704.44	1120.94	0.61627	0.87668	1.49295	441
1207.50	703.46	1121.03	0.61747	0.87462	1.49209	442
1207.60	702.48	1121.12	0.61867	0.87256	1.49123	443
1207.70	701.50	1121.21	0.61987	0.87050	1.49037	444
1207.80	700.53	1121.30	0.62106	0.86845	1.48951	445
1207.89	699.55	1121.38	0.62226	0.86640	1.48866	446
1207.98	698.56	1121.46	0.62346	0.86435	1.48781	447
1208.07	697.57	1121.54	0.62465	0.86230	1.48695	448
1208.16	696.59	1121.62	0.62585	0.86025	1.48610	449
1208.25	695.60	1121.70	0.62704	0.85821	1.48525	450
1208.33	694.61	1121.78	0.62823	0.85617	1.48440	451
1208.41	693.61	1121.85	0.62943	0.85413	1.48356	452
1208.49	692.62	1121.93	0.63062	0.85209	1.48271	453
1208.57	691.63	1122.00	0.63181	0.85006	1.48187	454
1208.64	690.63	1122.07	0.63300	0.84802	1.48102	455
1208.72	689.63	1122.15	0.63419	0.84599	1.48018	456
1208.79	688.63	1122.22	0.63537	0.84396	1.47933	457
1208.86	687.63	1122.29	0.63656	0.84193	1.47849	458
1208.93	686.63	1122.36	0.63775	0.83990	1.47765	459
1209.00	685.63	1122.43	0.63893	0.83787	1.47680	460
1209.06	684.62	1122.49	0.64011	0.83585	1.47596	461
1209.13	683.62	1122.56	0.64130	0.83382	1.47512	462
1209.19	682.61	1122.63	0.64248	0.83180	1.47428	463
1209.25	681.60	1122.69	0.64366	0.82978	1.47344	464
1209.31	680.59	1122.76	0.64484	0.82776	1.47260	465
1209.36	679.57	1122.81	0.64602	0.82574	1.47176	466
1209.41	678.56	1122.87	0.64720	0.82373	1.47093	467
1209.46	677.54	1122.92	0.64838	0.82171	1.47009	468
1209.51	676.52	1122.98	0.64956	0.81970	1.46926	469
1209.55	675.49	1123.03	0.65074	0.81769	1.46843	470
1209.60	674.47	1123.08	0.65192	0.81568	1.46760	471
1209.64	673.45	1123.13	0.65310	0.81367	1.46677	472
1209.68	672.42	1123.18	0.65427	0.81167	1.46594	473
1209.72	671.39	1123.23	0.65545	0.80966	1.46511	474
1209.75	670.36	1123.27	0.65662	0.80766	1.46428	475

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TABLE II. PRINCIPAL STEAM TABLE

0	1	2	3	4	5	6	7
t	p	s	s'	R/p	f_1	q	r
475	538.5	0.868	1.034	.001106	0.0938	454.9	754.9
476	543.7	0.859	1.025	.001096	0.0932	456.0	753.8
477	548.9	0.851	1.016	.001085	0.0926	457.1	752.7
478	554.1	0.843	1.008	.001075	0.0920	458.2	751.6
479	559.3	0.835	1.000	.001065	0.0915	459.3	750.6
480	564.6	0.827	0.991	.001055	0.0909	460.4	749.5
481	570.0	0.819	0.983	.001045	0.0903	461.5	748.4
482	575.3	0.811	0.975	.001035	0.0898	462.6	747.3
483	580.7	0.804	0.967	.001026	0.0892	463.7	746.2
484	586.2	0.796	0.959	.001016	0.0886	464.8	745.2
485	591.7	0.788	0.951	.001007	0.0881	465.9	744.1
486	597.2	0.781	0.943	.000997	0.0875	467.0	743.0
487	602.8	0.773	0.935	.000988	0.0870	468.1	741.9
488	608.4	0.766	0.928	.000979	0.0864	469.2	740.8
489	614.1	0.759	0.920	.000970	0.0859	470.3	739.7
490	619.8	0.752	0.913	.000961	0.0853	471.4	738.6
491	625.5	0.745	0.905	.000952	0.0848	472.6	737.4
492	631.3	0.738	0.898	.000944	0.0843	473.7	736.3
493	637.1	0.731	0.891	.000935	0.0837	474.8	735.2
494	643.0	0.724	0.883	.000926	0.0832	475.9	734.1
495	648.9	0.717	0.876	.000918	0.0827	477.0	733.0
496	654.8	0.710	0.869	.000910	0.0821	478.1	731.8
497	660.8	0.704	0.862	.000901	0.0816	479.2	730.7
498	666.8	0.697	0.855	.000893	0.0811	480.3	729.6
499	672.9	0.691	0.849	.000885	0.0805	481.4	728.4
500	679.0	0.684	0.842	.000877	0.0800	482.5	727.3
501	685.1	0.678	0.835	.000869	0.0795	483.6	726.1
502	691.3	0.671	0.828	.000862	0.0790	484.7	725.0
503	697.6	0.665	0.822	.000854	0.0785	485.9	723.8
504	703.9	0.659	0.815	.000846	0.0780	487.0	722.6
505	710.2	0.653	0.809	.000839	0.0775	488.1	721.5
506	716.6	0.647	0.803	.000831	0.0770	489.2	720.3
507	723.0	0.641	0.796	.000824	0.0765	490.3	719.1
508	729.5	0.635	0.790	.000816	0.0760	491.4	718.0
509	736.0	0.629	0.784	.000809	0.0755	492.5	716.8
510	742.5	0.623	0.778	.000802	0.0751	493.6	715.6
511	749.2	0.617	0.772	.000795	0.0746	494.8	714.4
512	755.8	0.611	0.766	.000788	0.0741	495.9	713.2
513	762.5	0.606	0.760	.000781	0.0737	497.0	712.0
514	769.2	0.600	0.754	.000774	0.0732	498.1	710.8
515	776.0	0.595	0.748	.000768	0.0727	499.2	709.6
516	782.9	0.589	0.742	.000761	0.0723	500.4	708.3
517	789.7	0.584	0.737	.000754	0.0718	501.5	707.1
518	796.7	0.578	0.731	.000748	0.0714	502.6	705.9
519	803.6	0.573	0.725	.000741	0.0709	503.7	704.7
520	810.7	0.568	0.720	.000735	0.0705	504.9	703.4
521	817.7	0.562	0.714	.000728	0.0700	506.0	702.2
522	824.8	0.557	0.709	.000722	0.0696	507.1	700.9
523	832.0	0.552	0.704	.000716	0.0692	508.2	699.7
524	839.2	0.547	0.698	.000710	0.0687	509.4	698.4
525	846.5	0.542	0.693	.000704	0.0683	510.5	697.1

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PRINCIPAL STEAM TABLE TABLE II.

8	9	10	11	12	13	0
H	l	K	a	b	N	t
1209.8	670.4	1123.3	0.6566	0.8077	1.4643	475
1209.8	669.3	1123.3	0.6578	0.8057	1.4635	476
1209.8	668.3	1123.4	0.6590	0.8037	1.4627	477
1209.8	667.2	1123.4	0.6601	0.8017	1.4618	478
1209.9	666.2	1123.4	0.6613	0.7997	1.4610	479
1209.9	665.2	1123.5	0.6625	0.7977	1.4602	480
1209.9	664.1	1123.5	0.6636	0.7957	1.4593	481
1209.9	663.1	1123.5	0.6648	0.7937	1.4585	482
1209.9	662.0	1123.6	0.6660	0.7917	1.4577	483
1210.0	661.0	1123.6	0.6671	0.7897	1.4568	484
1210.0	659.9	1123.6	0.6683	0.7877	1.4560	485
1210.0	658.9	1123.7	0.6695	0.7857	1.4552	486
1210.0	657.8	1123.7	0.6707	0.7837	1.4544	487
1210.0	656.8	1123.7	0.6718	0.7817	1.4535	488
1210.0	655.7	1123.7	0.6730	0.7797	1.4527	489
1210.0	654.7	1123.8	0.6741	0.7778	1.4519	490
1210.0	653.6	1123.8	0.6753	0.7758	1.4511	491
1210.0	652.5	1123.8	0.6765	0.7738	1.4503	492
1210.0	651.5	1123.8	0.6776	0.7718	1.4494	493
1210.0	650.4	1123.8	0.6788	0.7698	1.4486	494
1209.9	649.3	1123.8	0.6800	0.7678	1.4478	495
1209.9	648.2	1123.8	0.6811	0.7658	1.4469	496
1209.9	647.2	1123.8	0.6823	0.7638	1.4461	497
1209.9	646.1	1123.8	0.6834	0.7619	1.4453	498
1209.8	645.0	1123.8	0.6846	0.7599	1.4445	499
1209.8	643.9	1123.8	0.6858	0.7579	1.4437	500
1209.7	642.8	1123.8	0.6869	0.7559	1.4428	501
1209.7	641.7	1123.8	0.6881	0.7539	1.4420	502
1209.7	640.6	1123.8	0.6892	0.7520	1.4412	503
1209.6	639.5	1123.8	0.6904	0.7500	1.4404	504
1209.6	638.4	1123.8	0.6915	0.7480	1.4395	505
1209.5	637.3	1123.7	0.6927	0.7460	1.4387	506
1209.4	636.2	1123.7	0.6938	0.7440	1.4378	507
1209.4	635.1	1123.7	0.6950	0.7420	1.4370	508
1209.3	634.0	1123.7	0.6962	0.7400	1.4362	509
1209.2	632.8	1123.6	0.6973	0.7380	1.4353	510
1209.2	631.7	1123.6	0.6985	0.7360	1.4345	511
1209.1	630.6	1123.6	0.6996	0.7341	1.4337	512
1209.0	629.5	1123.5	0.7008	0.7321	1.4329	513
1208.9	628.3	1123.5	0.7019	0.7301	1.4320	514
1208.8	627.2	1123.4	0.7031	0.7281	1.4312	515
1208.7	626.0	1123.4	0.7042	0.7261	1.4303	516
1208.6	624.9	1123.3	0.7054	0.7241	1.4295	517
1208.5	623.7	1123.3	0.7065	0.7221	1.4286	518
1208.4	622.6	1123.2	0.7077	0.7201	1.4278	519
1208.3	621.4	1123.1	0.7088	0.7181	1.4269	520
1208.2	620.3	1123.1	0.7099	0.7161	1.4260	521
1208.0	619.1	1123.0	0.7111	0.7141	1.4252	522
1207.9	617.9	1122.9	0.7122	0.7121	1.4243	523
1207.8	616.7	1122.8	0.7134	0.7100	1.4234	524
1207.6	615.6	1122.7	0.7146	0.7080	1.4226	525

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TABLE II. PRINCIPAL STEAM TABLE

0	1	2	3	4	5	6	7
t	p	s	s'	R/p	f_1	q	r
525	846.5	0.542	0.693	.000704	0.0683	510.5	697.1
526	853.8	0.537	0.688	.000698	0.0679	511.6	695.9
527	861.2	0.532	0.682	.000692	0.0674	512.7	694.6
528	868.6	0.527	0.677	.000686	0.0670	513.9	693.3
529	876.1	0.522	0.672	.000680	0.0666	515.0	692.0
530	883.6	0.517	0.667	.000674	0.0662	516.1	690.7
531	891.2	0.513	0.662	.000668	0.0657	517.3	689.4
532	898.8	0.508	0.657	.000663	0.0653	518.4	688.1
533	906.5	0.503	0.652	.000657	0.0649	519.5	686.8
534	914.2	0.499	0.647	.000652	0.0645	520.7	685.4
535	922.0	0.494	0.643	.000646	0.0641	521.8	684.1
536	929.8	0.490	0.638	.000641	0.0637	523.0	682.7
537	937.7	0.485	0.633	.000635	0.0632	524.1	681.4
538	945.6	0.481	0.628	.000630	0.0628	525.2	680.0
539	953.6	0.476	0.624	.000625	0.0624	526.4	678.6
540	961.6	0.472	0.619	.000619	0.0620	527.5	677.3
541	969.7	0.468	0.615	.000614	0.0616	528.6	675.9
542	977.9	0.463	0.610	.000609	0.0612	529.8	674.5
543	986.1	0.459	0.606	.000604	0.0608	530.9	673.1
544	994.3	0.455	0.601	.000599	0.0604	532.1	671.6
545	1002.6	0.451	0.597	.000594	0.0600	533.2	670.2
546	1011.0	0.447	0.593	.000589	0.0596	534.4	668.8
547	1019.4	0.443	0.588	.000584	0.0592	535.5	667.4
548	1027.9	0.438	0.584	.000580	0.0589	536.7	665.9
549	1036.4	0.434	0.580	.000575	0.0585	537.8	664.4
550	1045.0	0.430	0.575	.000570	0.0581	538.9	663.0
560	1133.9	0.393	0.536	.000522	0.0542	550.5	647.6
570	1228.7	0.358	0.499	.000485	0.0506	562.1	631.2
580	1329.6	0.326	0.465	.000448	0.0472	573.9	613.3
590	1436.9	0.297	0.435	.000415	0.0440	585.8	593.8
600	1550.7	0.270	0.407	.000384	0.0410	598.0	572.4
610	1672	0.245	0.381		0.0382	610.4	548.8
620	1800	0.221	0.357		0.0355	623.1	522.3
630	1935	0.199	0.335		0.0330	636.3	492.4
640	2078	0.178	0.315		0.0308	650.4	458.2
650	2229	0.157	0.298		0.0288	666.0	418.6
660	2388	0.137	0.279		0.0270	684	372
670	2556	0.115	0.262		0.0254	707	311
680	2733	0.092	0.248		0.0239	740	229
689	2900	0.049	0.236		0.0226	847	0

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PRINCIPAL STEAM TABLE TABLE II.

8	9	10	11	12	13	0
H	l	K	a	b	N	t
1207.6	615.6	1122.7	0.7146	0.7080	1.4226	525
1207.5	614.4	1122.6	0.7157	0.7060	1.4217	526
1207.3	613.2	1122.5	0.7168	0.7040	1.4208	527
1207.2	612.0	1122.4	0.7180	0.7020	1.4200	528
1207.0	610.8	1122.3	0.7191	0.7000	1.4191	529
1206.8	609.6	1122.2	0.7203	0.6979	1.4182	530
1206.7	608.3	1122.1	0.7214	0.6959	1.4173	531
1206.5	607.1	1122.0	0.7226	0.6939	1.4165	532
1206.3	605.9	1121.8	0.7237	0.6919	1.4156	533
1206.1	604.7	1121.7	0.7249	0.6898	1.4147	534
1205.9	603.4	1121.6	0.7260	0.6878	1.4138	535
1205.7	602.2	1121.4	0.7271	0.6858	1.4129	536
1205.5	600.9	1121.3	0.7283	0.6837	1.4120	537
1205.2	599.7	1121.1	0.7294	0.6817	1.4111	538
1205.0	598.4	1121.0	0.7305	0.6796	1.4101	539
1204.8	597.1	1120.8	0.7317	0.6775	1.4092	540
1204.5	595.8	1120.6	0.7328	0.6755	1.4083	541
1204.3	594.5	1120.4	0.7340	0.6734	1.4074	542
1204.0	593.2	1120.2	0.7351	0.6713	1.4064	543
1203.7	591.9	1120.0	0.7363	0.6692	1.4055	544
1203.4	590.6	1119.8	0.7374	0.6671	1.4045	545
1203.2	589.3	1119.6	0.7385	0.6651	1.4036	546
1202.9	588.0	1119.4	0.7397	0.6630	1.4026	547
1202.6	586.7	1119.2	0.7408	0.6609	1.4017	548
1202.2	585.3	1118.9	0.7419	0.6588	1.4007	549
1201.9	583.9	1118.7	0.7431	0.6566	1.3997	550
1198.1	569.9	1115.7	0.7545	0.6352	1.3897	560
1193.3	554.9	1111.9	0.7658	0.6131	1.3789	570
1187.2	538.6	1106.9	0.7772	0.5899	1.3671	580
1179.6	521.0	1100.6	0.7886	0.5657	1.3543	590
1170.4	501.7	1092.9	0.8001	0.5402	1.3403	600
1159.2	480.6	1083.4	0.8117	0.5131	1.3248	610
1145.4	456.9	1071.7	0.8235	0.4838	1.3073	620
1128.7	430.3	1057.5	0.8357	0.4519	1.2876	630
1108.6	400.0	1040.3	0.8486	0.4167	1.2653	640
1084.6	365.1	1019.9	0.8627	0.3772	1.2399	650
1056	324	995	0.879	0.332	1.211	660
1018	271	964	0.900	0.275	1.175	670
969	200	923	0.929	0.201	1.130	680
847	0	821	1.022	0.0	1.022	689

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TABLE III. SUPPLEMENTARY STEAM TABLE

0	1	2	3	4	5	6	7	8
t	p	w	d_w	pw	ps	c_w	APw	APu
32	0.089	.01602	62.42	0.001	292.68	1.0067	.0002	54.17
40	0.122	.01602	62.43	0.002	297.41	1.0033	.0004	55.05
50	0.178	.01602	62.42	0.003	303.31	1.0000	.0005	56.14
60	0.256	.01603	62.37	0.004	309.20	0.9977	.0007	57.23
70	0.363	.01605	62.30	0.006	315.06	0.9966	.0011	58.31
80	0.506	.01607	62.21	0.008	320.90	0.9963	.0015	59.40
90	0.696	.01610	62.11	0.011	326.72	0.9964	.0020	60.47
100	0.946	.01613	62.00	0.015	332.50	0.9966	.0028	61.54
110	1.271	.01616	61.86	0.021	338.24	0.9969	.0039	62.60
120	1.689	.01620	61.71	0.027	343.94	0.9974	.0052	63.65
130	2.219	.01625	61.55	0.036	349.60	0.9980	.0067	64.70
140	2.885	.01629	61.38	0.047	355.21	0.9988	.0087	65.74
150	3.715	.01634	61.19	0.061	360.76	0.9996	.0113	66.76
160	4.74	.01639	60.99	0.078	366.25	1.0006	.0144	67.78
170	5.99	.01645	60.79	0.099	371.67	1.0018	.0183	68.77
180	7.51	.01651	60.57	0.124	377.03	1.0031	.0229	69.76
190	9.34	.01657	60.35	0.155	382.30	1.0044	.0285	70.73
200	11.53	.01663	60.12	0.192	387.49	1.0059	.0353	71.68
210	14.12	.01670	59.88	0.236	392.59	1.0076	0.044	72.62
220	17.19	.01677	59.63	0.288	397.60	1.0093	0.053	73.54
230	20.78	.01684	59.37	0.350	402.50	1.0112	0.065	74.43
240	24.97	.01692	59.11	0.423	407.28	1.0133	0.078	75.30
250	29.82	.01700	58.83	0.507	411.92	1.0151	0.094	76.15
260	35.42	.01708	58.55	0.605	416.42	1.0177	0.112	76.96
270	41.85	.01716	58.26	0.719	420.78	1.0201	0.133	77.75
280	49.19	.01725	57.96	0.850	425.00	1.0226	0.157	78.50
290	57.34	.01735	57.65	0.999	429.06	1.0253	0.185	79.23
300	66.98	.01744	57.33	1.168	432.97	1.0281	0.216	79.92
310	77.6	.01754	57.00	1.362	436.71	1.0310	0.252	80.58
320	89.6	.01765	56.66	1.581	440.28	1.0340	0.293	81.20
330	103.0	.01776	56.31	1.830	443.68	1.0372	0.339	81.78
340	117.9	.01788	55.95	2.108	446.90	1.0405	0.390	82.33
350	134.5	.01800	55.57	2.421	449.93	1.0439	0.448	82.83
360	152.9	.01812	55.18	2.77	452.76	1.0475	0.512	83.29
370	173.2	.01825	54.78	3.16	455.40	1.0512	0.585	83.70
380	195.5	.01839	54.37	3.60	457.82	1.0550	0.666	84.07
390	220.1	.01854	53.95	4.08	460.01	1.0589	0.755	84.38
400	246.9	.01869	53.52	4.62	461.97	1.0630	0.855	84.65
410	276.3	.01884	53.08	5.21	463.67	1.0672	0.965	84.85
420	308.3	.01900	52.63	5.86	465.10	1.0715	1.084	85.00
430	343.1	.01917	52.17	6.58	466.24	1.0759	1.216	85.08
440	380.8	.01935	51.70	7.37	467.07	1.0805	1.363	85.09
450	421.7	.01953	51.22	8.23	467.57	1.0852	1.523	85.02
460	465.8	.01971	50.74	9.18	467.72	1.0900	1.699	84.87
470	513.4	.01990	50.25	10.21	467.50	1.0950	1.890	84.64
480	564.6	.02010	49.74	11.33	466.90	1.1002	2.098	84.32
490	619.8	.02031	49.22	12.59	465.88	1.1056	2.329	83.90
500	679.0	.02053	48.68	13.95	464.41	1.1113	2.582	83.38

For definitions and explanations of quantities, see pages 574 to 577.

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TABLE III — *Continued.*

0	1	2	3	4	5	6	7	8
t	p	w	d_w	pw	ps	c_w	APw	APu
510	743	.02077	48.12	15.43	462.5	1.1175	2.86	82.76
520	811	.02103	47.54	17.05	460.1	1.1244	3.16	82.00
530	884	.02131	46.94	18.83	457.2	1.1320	3.48	81.13
540	962	.02161	46.31	20.77	453.7	1.140	3.84	80.14
550	1045	.02192	45.66	22.89	449.8	1.149	4.23	79.02
560	1134	.02224	44.99	25.2	445.3	1.159	4.66	77.75
570	1229	.02257	44.29	27.8	440.0	1.170	5.13	76.31
580	1330	.02294	43.56	30.5	433.9	1.185	5.65	74.67
590	1437	.02339	42.79	33.6	426.9	1.205	6.22	72.82
600	1551	.02383	41.97	37.0	418.9	1.245	6.85	70.67
610	1672	.02434	41.09	40.7	409.4		7.54	68.23
620	1800	.02491	40.14	44.8	398.2		8.29	65.40
630	1935	.02558	39.10	49.5	385.0		9.16	62.09
640	2078	.02635	37.95	54.8	369.2		10.14	58.20
650	2229	.02727	36.67	60.8	350.0		11.26	53.52
660	2388	.0284	35.2	67.8	326.3		12.6	47.8
670	2556	.0299	33.5	76.8	294.2		14.2	40.2
680	2733	.0324	30.9	90.0	250.2		16.7	29.6
689	2900	.0487	20.5	141.3	141.3		26.2	0

TABLE IV. TEMPERATURE FACTOR f_t FOR HIGH RANGE, ABOVE COLUMN 5 OF TABLE II. SEE PAGE 72.

t	600	700	800	900	1000	1100	1200
0	.0410	.0212	.0119	.0068	.0034	.0017	.0009
10	.0382	.0200	.0112	.0064	.0032	.0016	.0009
20	.0355	.0188	.0106	.0059	.0030	.0015	.0008
30	.0330	.0177	.0100	.0055	.0028	.0014	.0008
40	.0308	.0167	.0095	.0052	.0027	.0014	.0007
50	.0288	.0158	.0090	.0048	.0025	.0013	.0007
60	.0270	.0149	.0086	.0045	.0023	.0012	.0007
70	.0254	.0141	.0081	.0042	.0021	.0011	.0006
80	.0239	.0133	.0077	.0039	.0020	.0010	.0006
90	.0225	.0126	.0072	.0036	.0018	.0010	.0006
100	.0212	.0119	.0068	.0034	.0017	.0009	.0005

TABLE V. PRESSURE FACTOR f_p FOR HIGH RANGE, ABOVE EQUATION (65). SEE PAGE 72.

p	f_p	Δf_p	p	f_p	Δf_p	p	f_p	Δf_p
900	2.2859	.0002	1100	2.5817	.0103	1350	2.9836	.0550
950	2.3589	.0018	1150	2.6587	.0158	1400	3.0686	.0686
1000	2.4322	.0036	1200	2.7371	.0228	1450	3.155	.084
1050	2.5062	.0062	1250	2.8174	.0317	1500	3.245	.102
			1300	2.8991	.0420			

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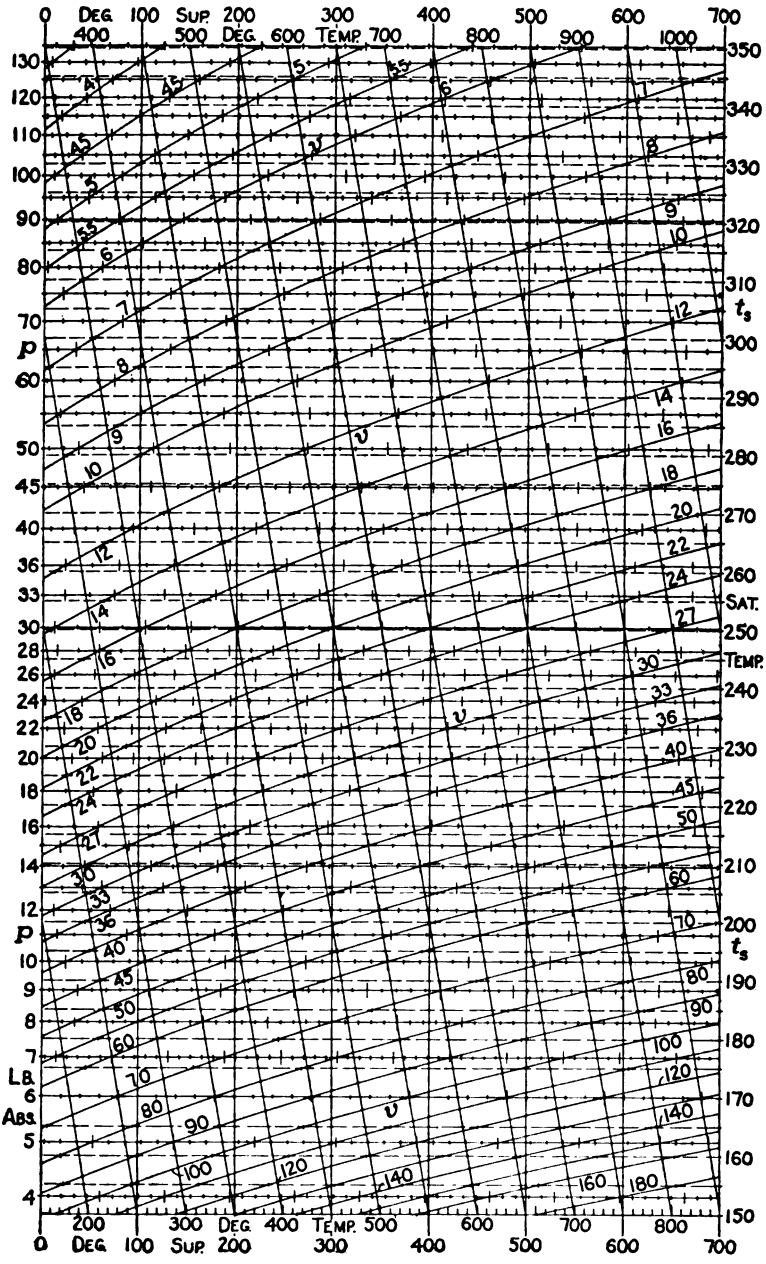

TABLE VI, page A. SPECIFIC VOLUME OF

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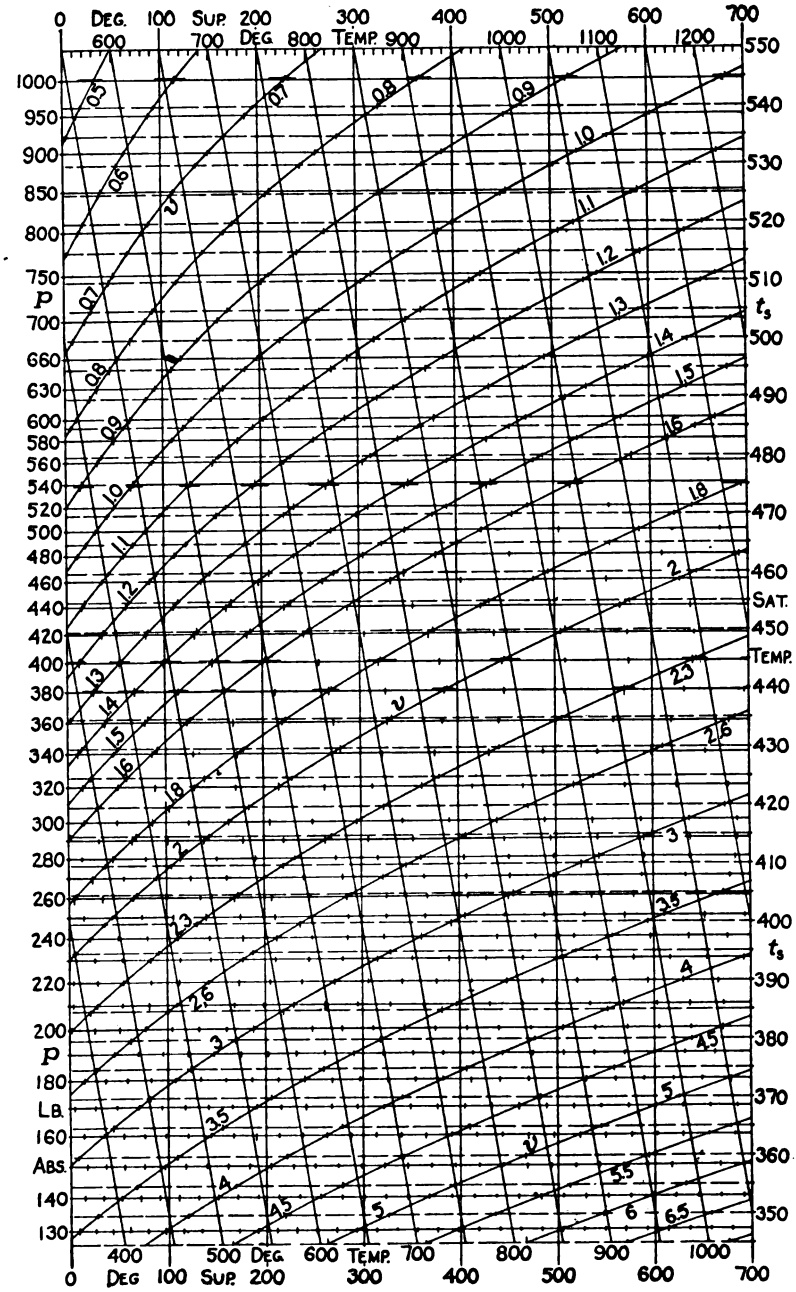

SUPERHEATED STEAM. TABLE VI, page B.

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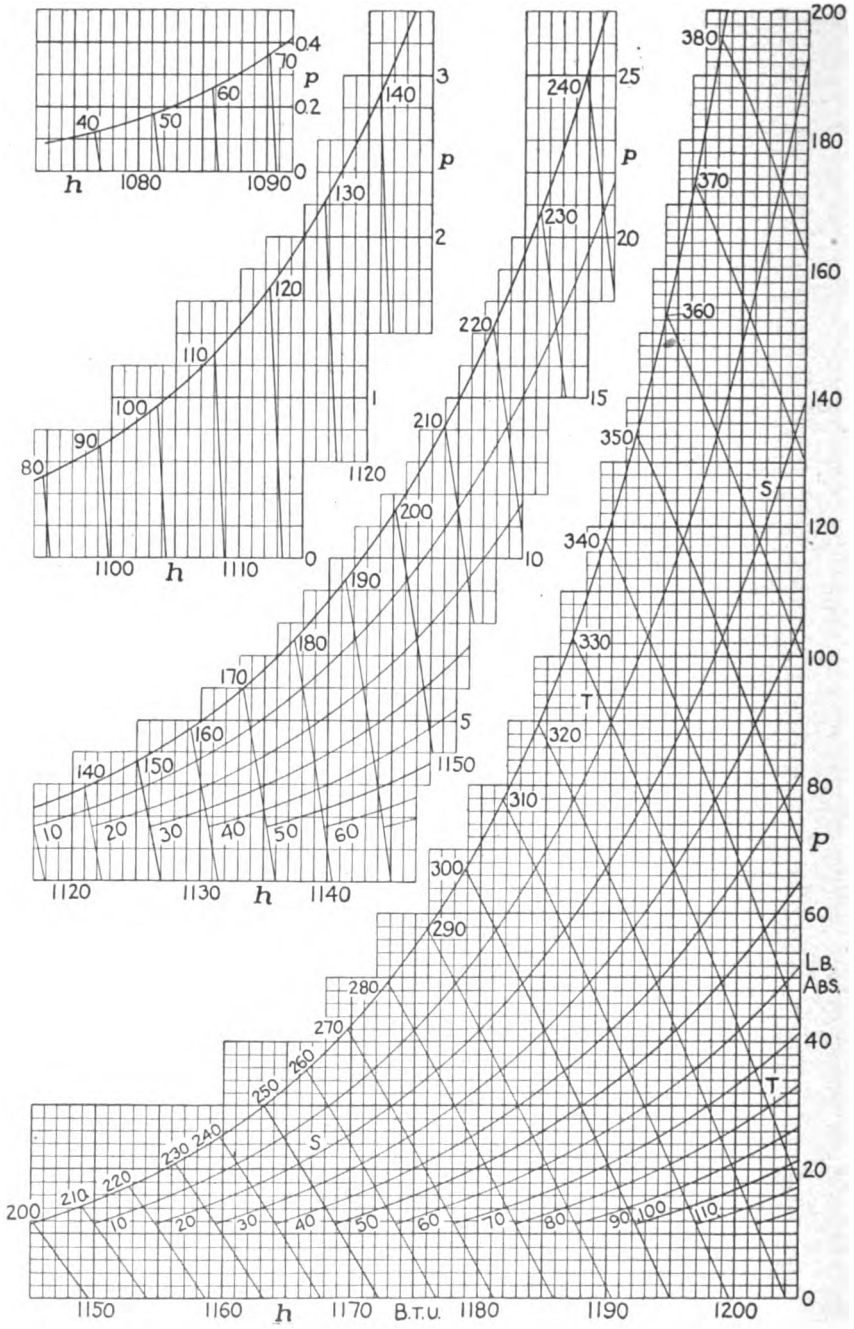
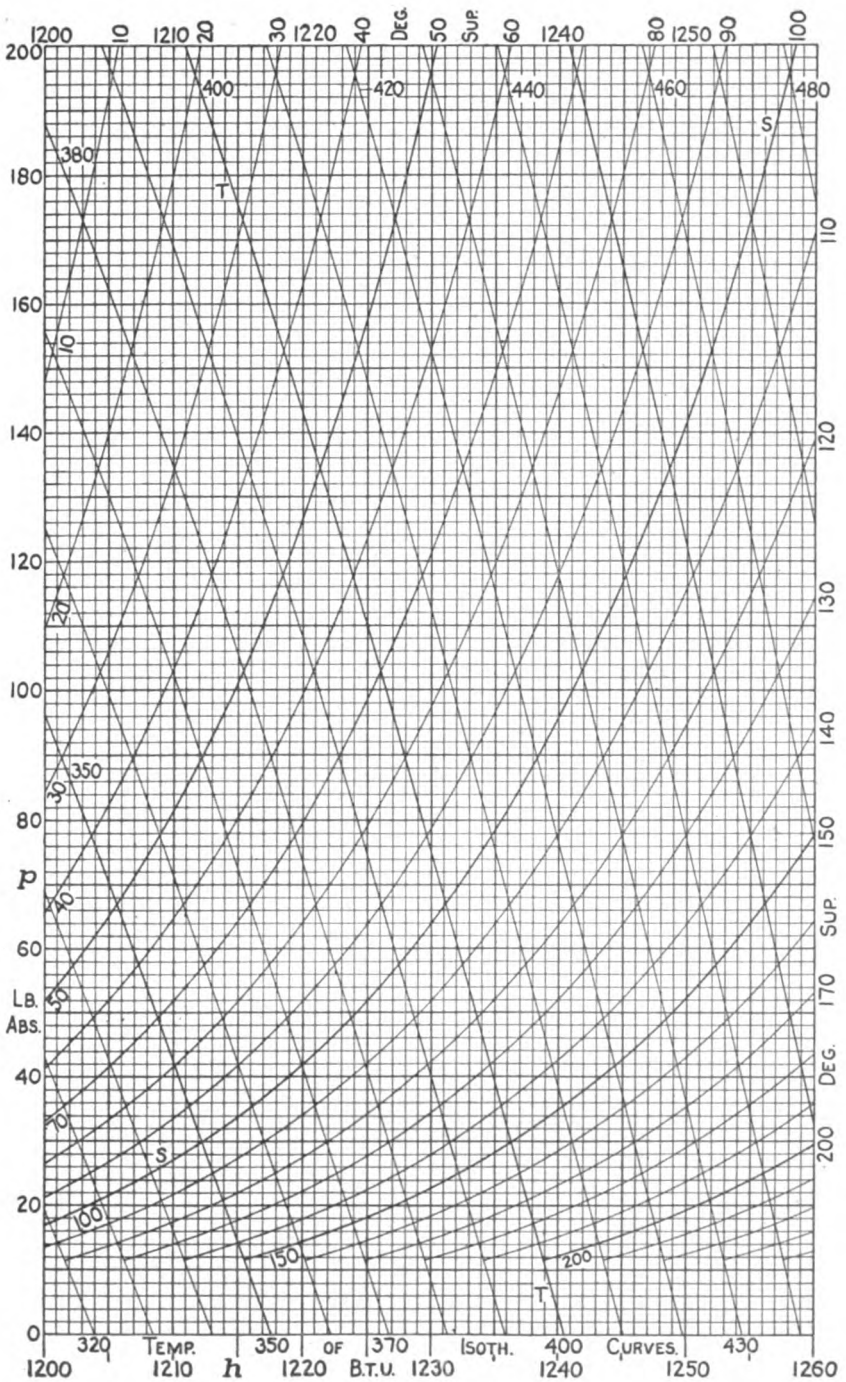


TABLE VII, page A. TOTAL HEAT OF



SUPERHEATED STEAM. TABLE VII, page B.

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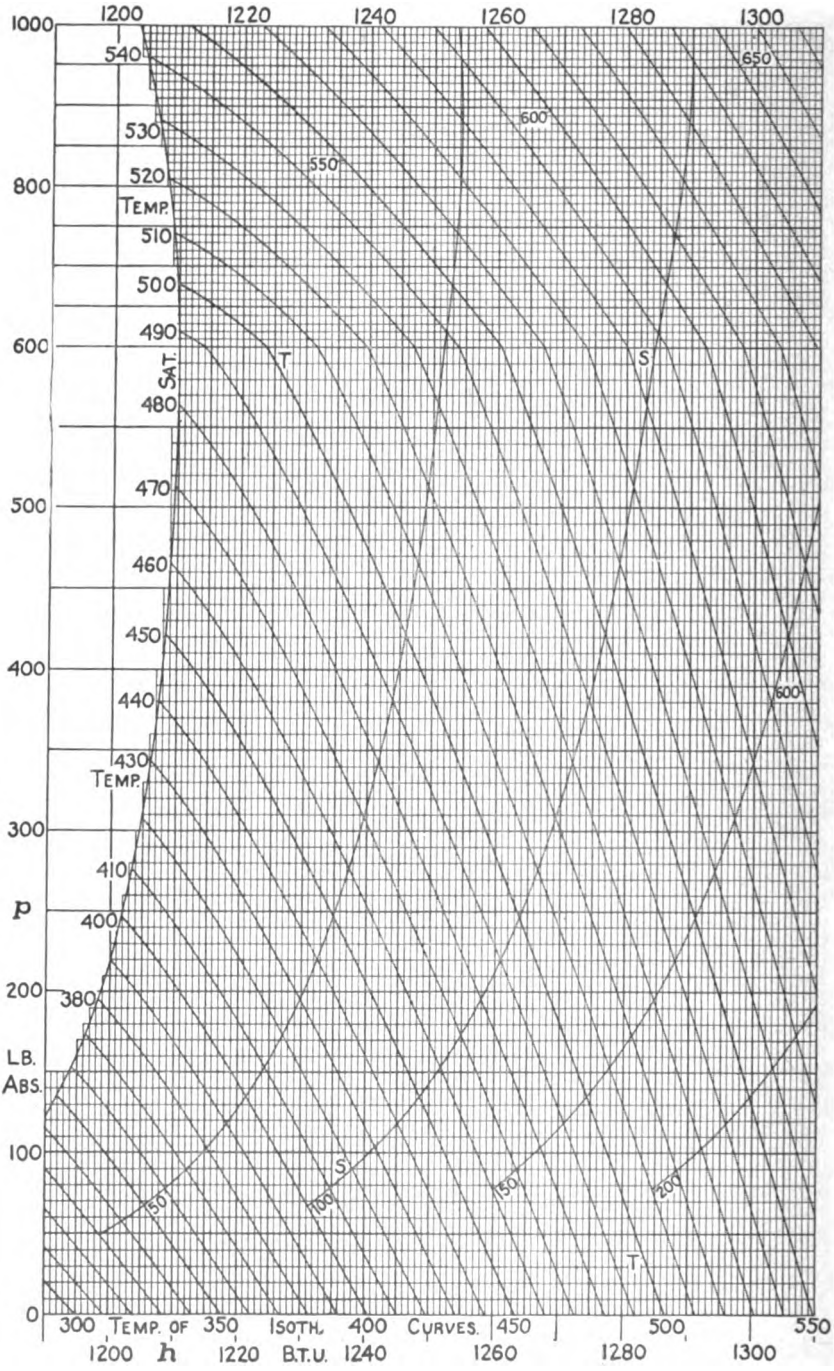

TABLE VII, page C.

TOTAL HEAT OF

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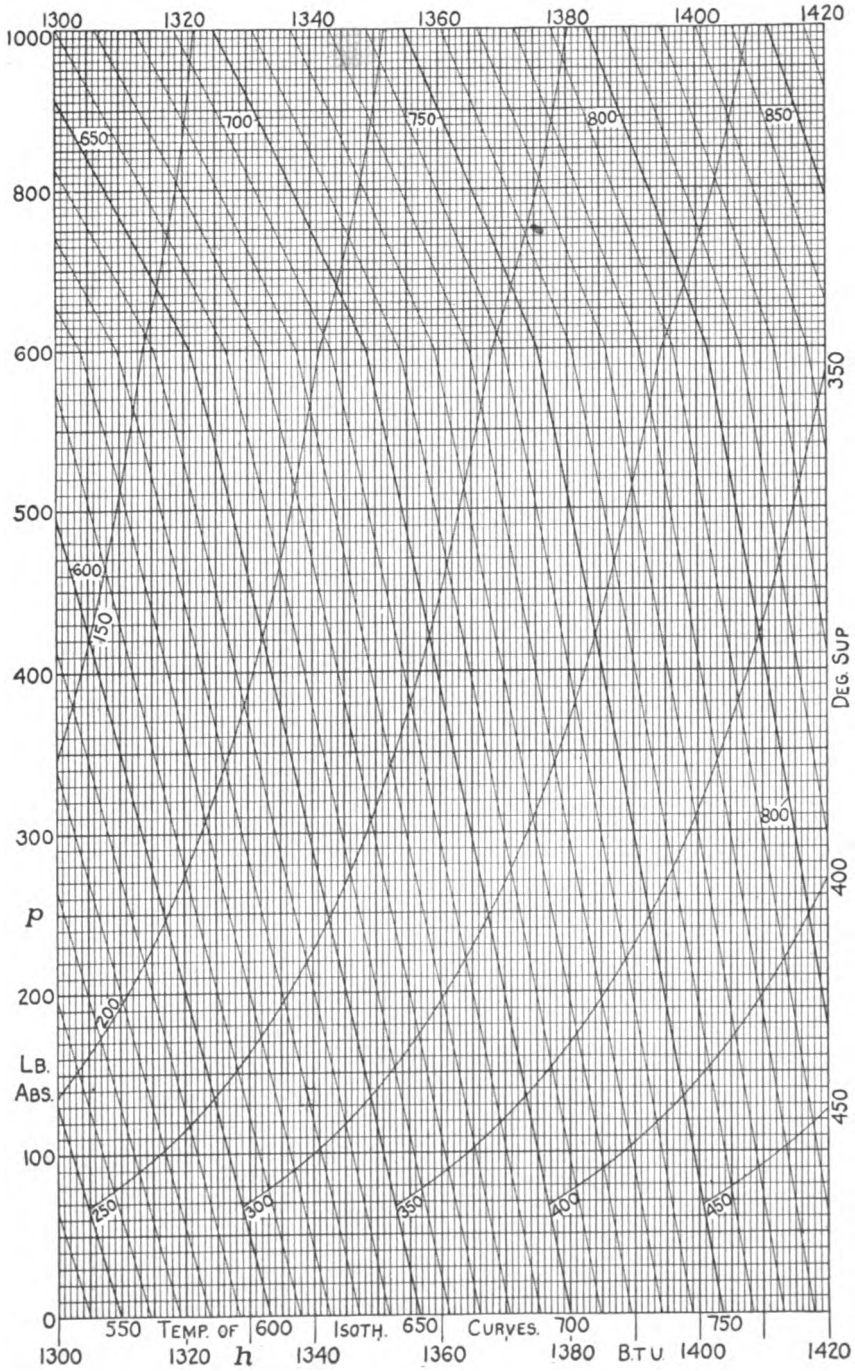

SUPERHEATED STEAM.

TABLE VII, page D.

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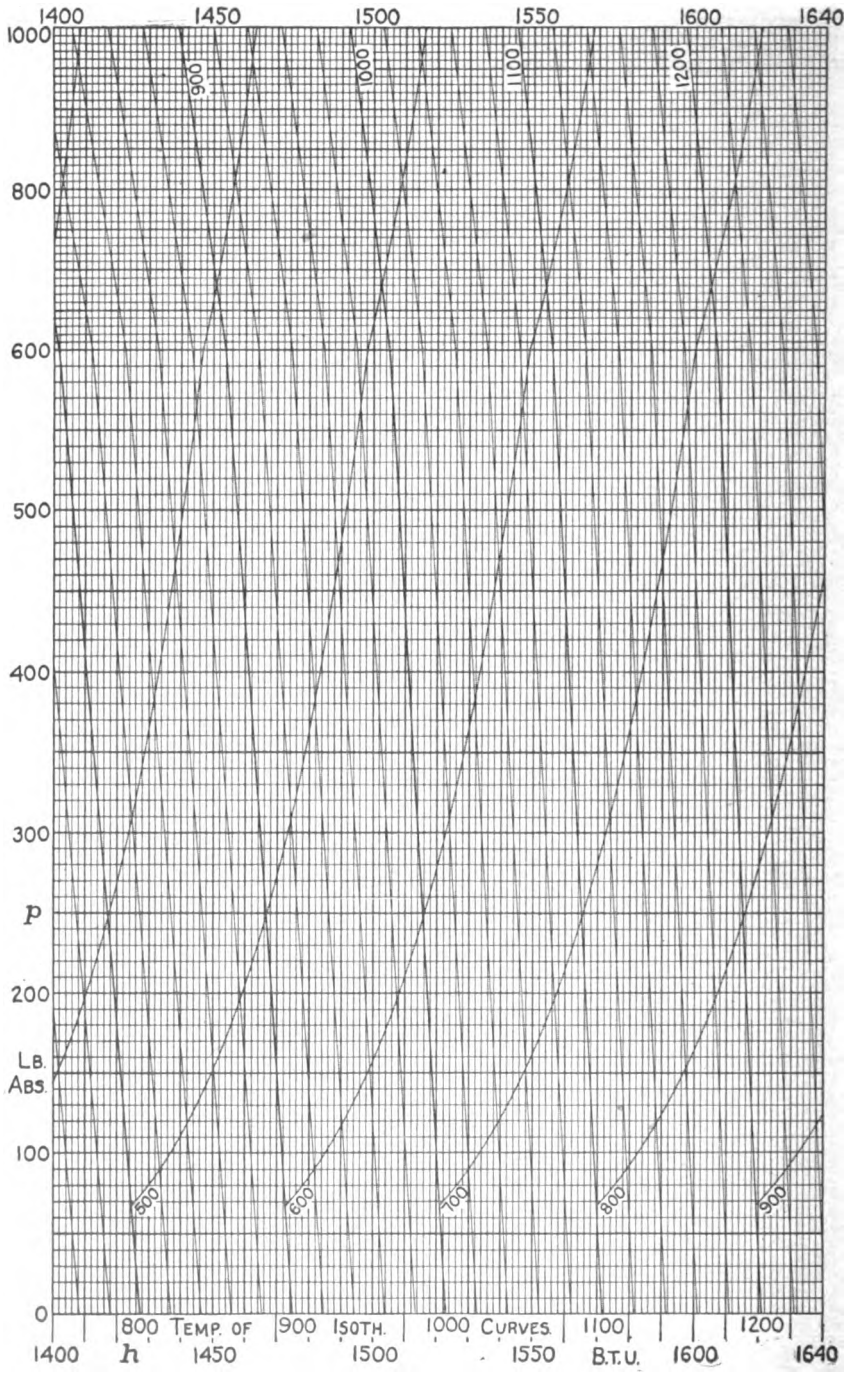

TABLE VII, page E. TOTAL HEAT.

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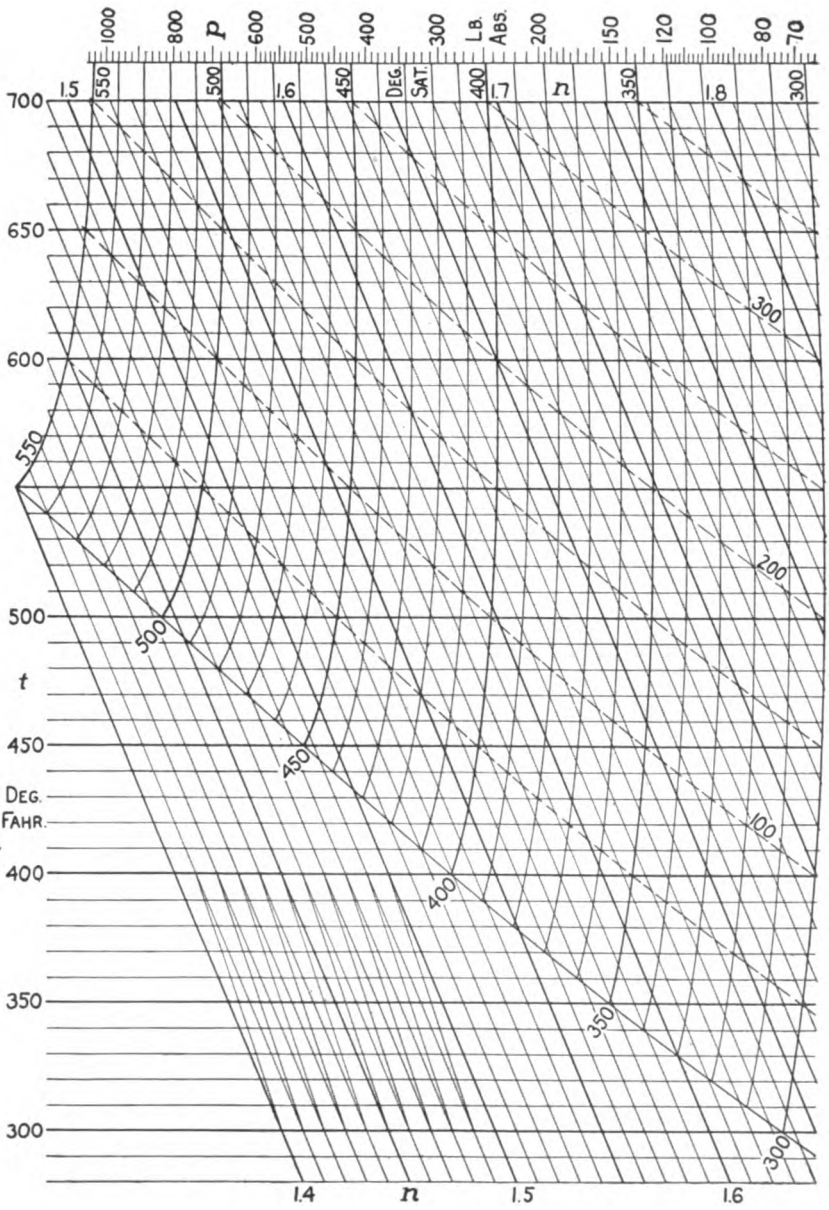

TABLE VIII, page A.

ENTROPY OF SUPERHEATED STEAM.

Pages A, B, and C run side by side in sequence, pages D and E go above them.

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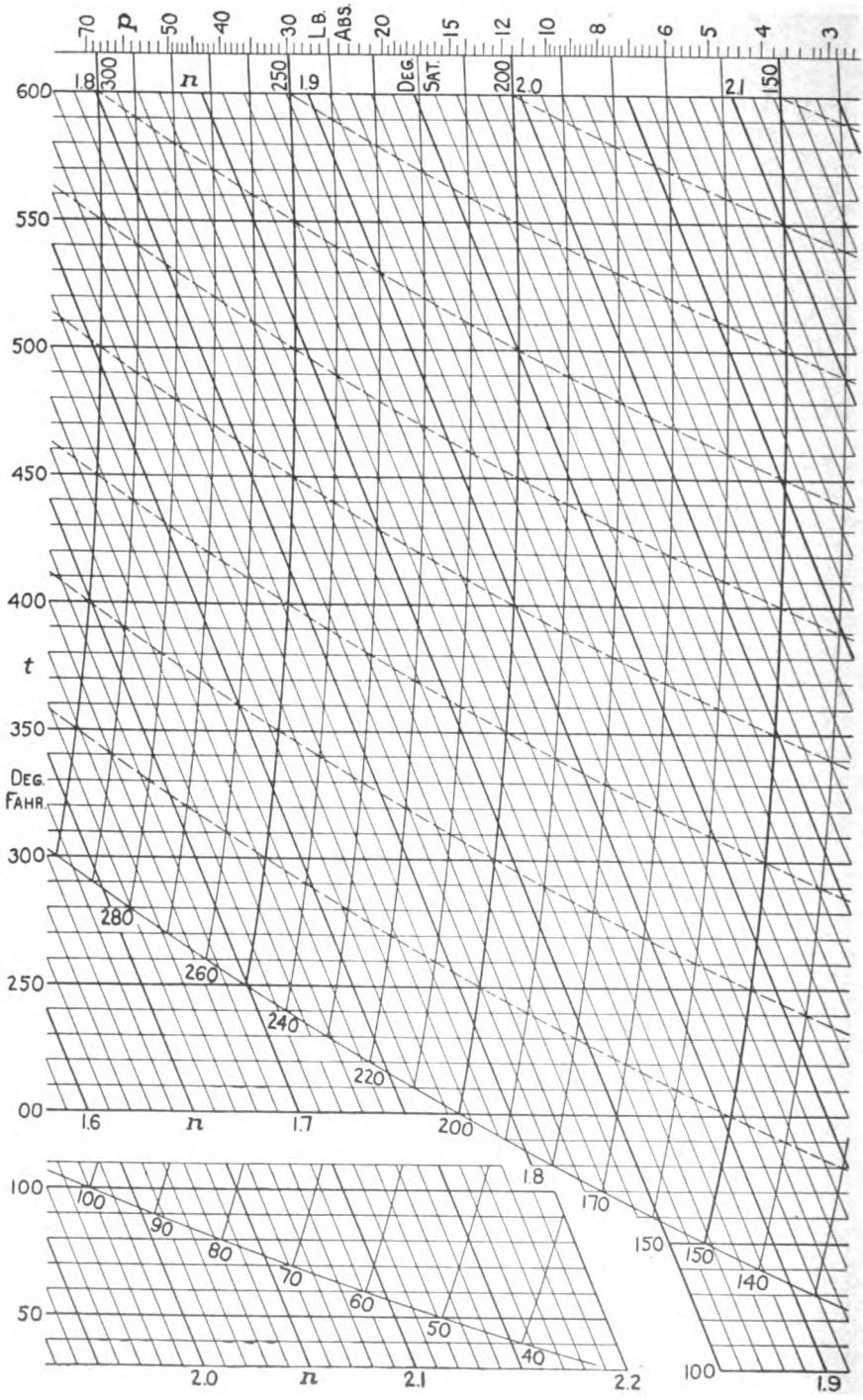

TABLE VIII, page B. ENTROPY OF

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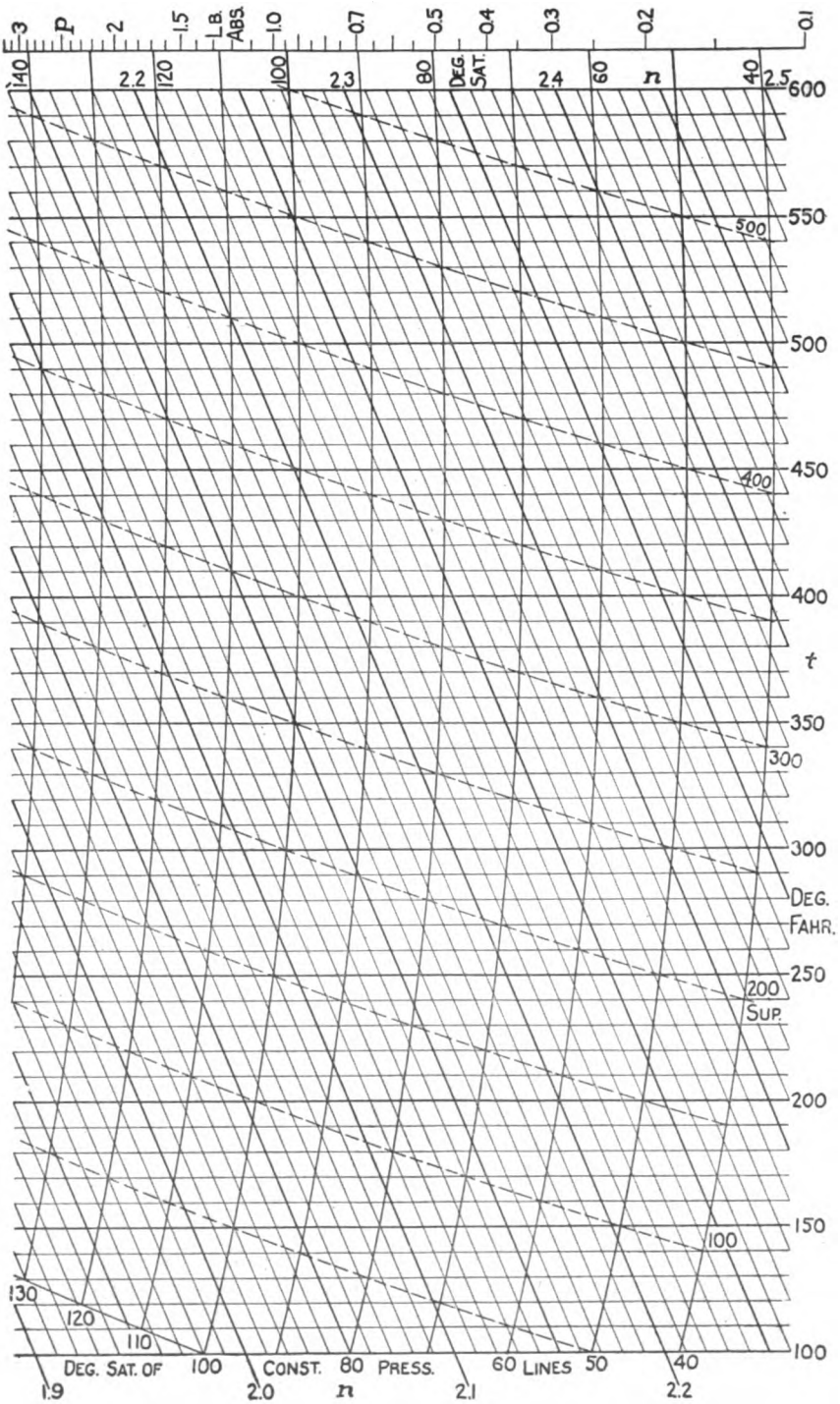

SUPERHEATED STEAM.

TABLE VIII, page C.

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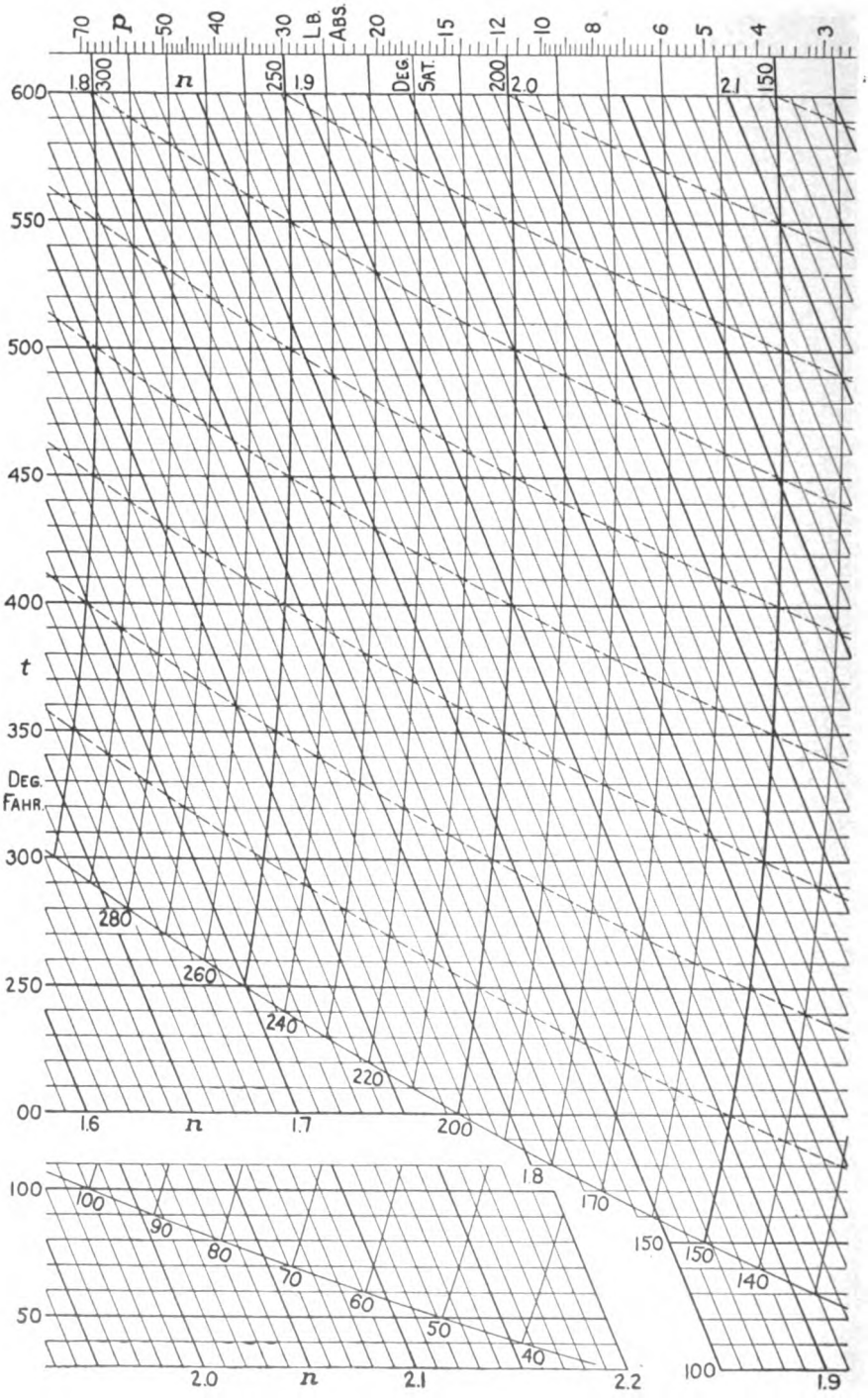

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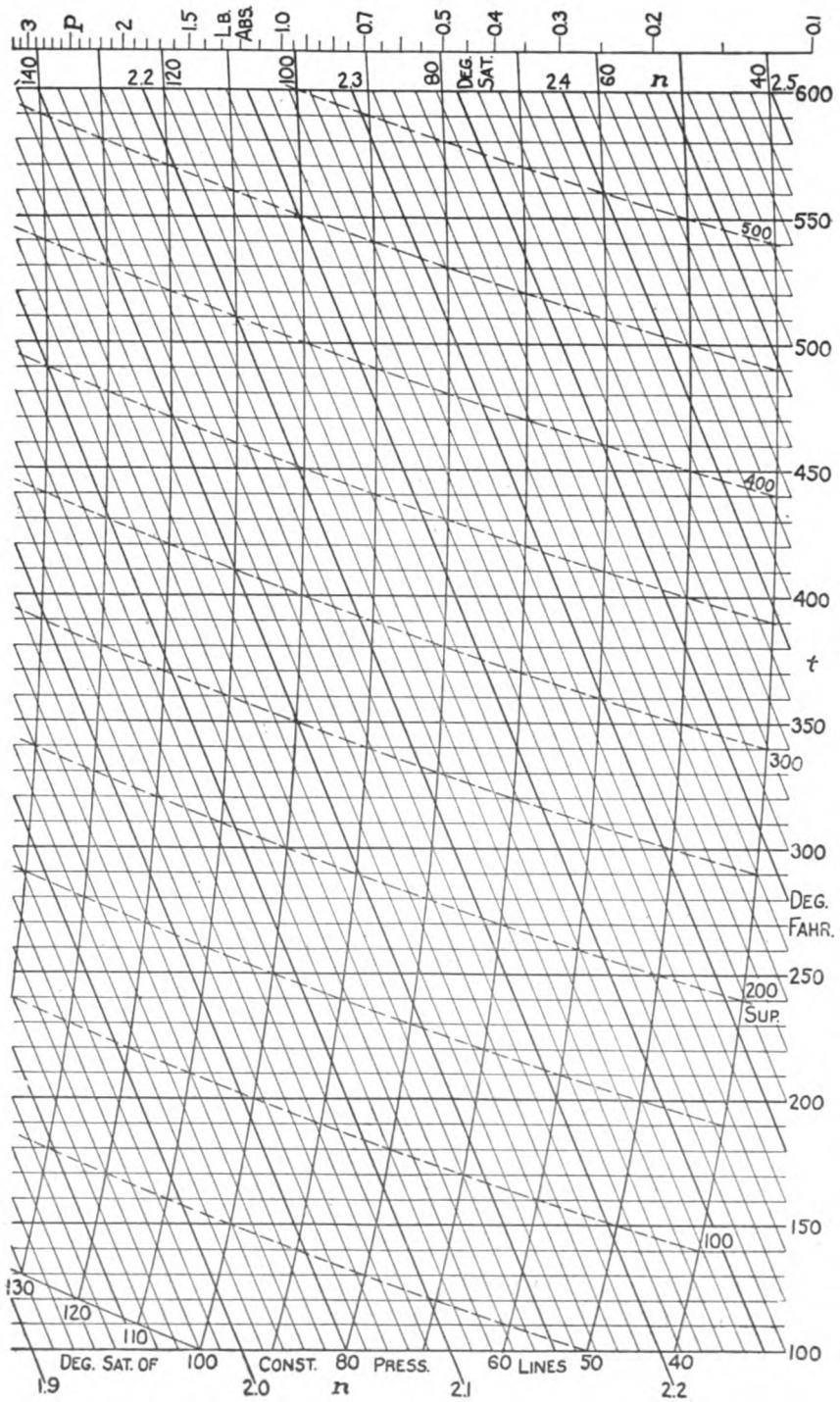

SUPERHEATED STEAM. TABLE VIII, page C.

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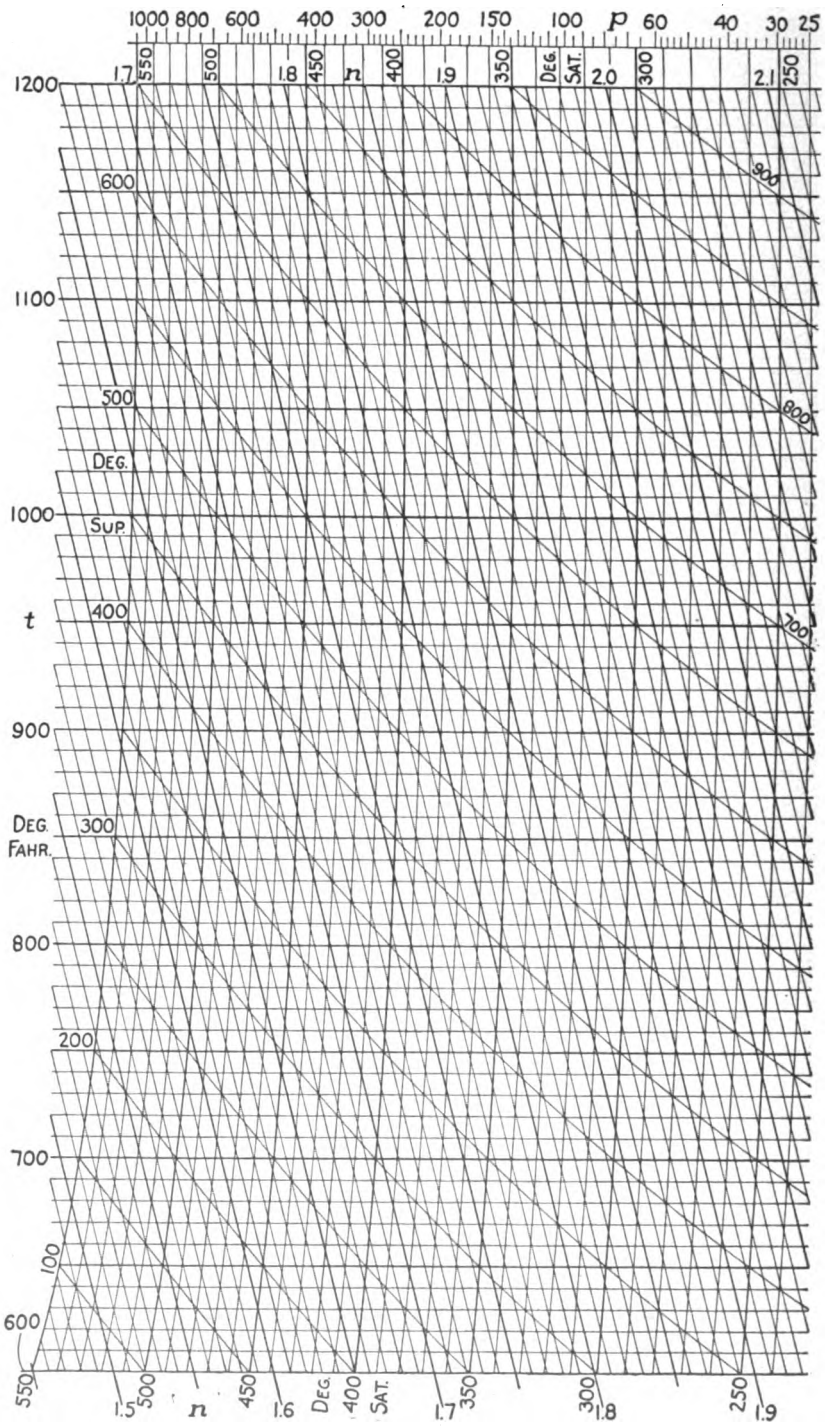

TABLE VIII, page D.

ENTROPY OF

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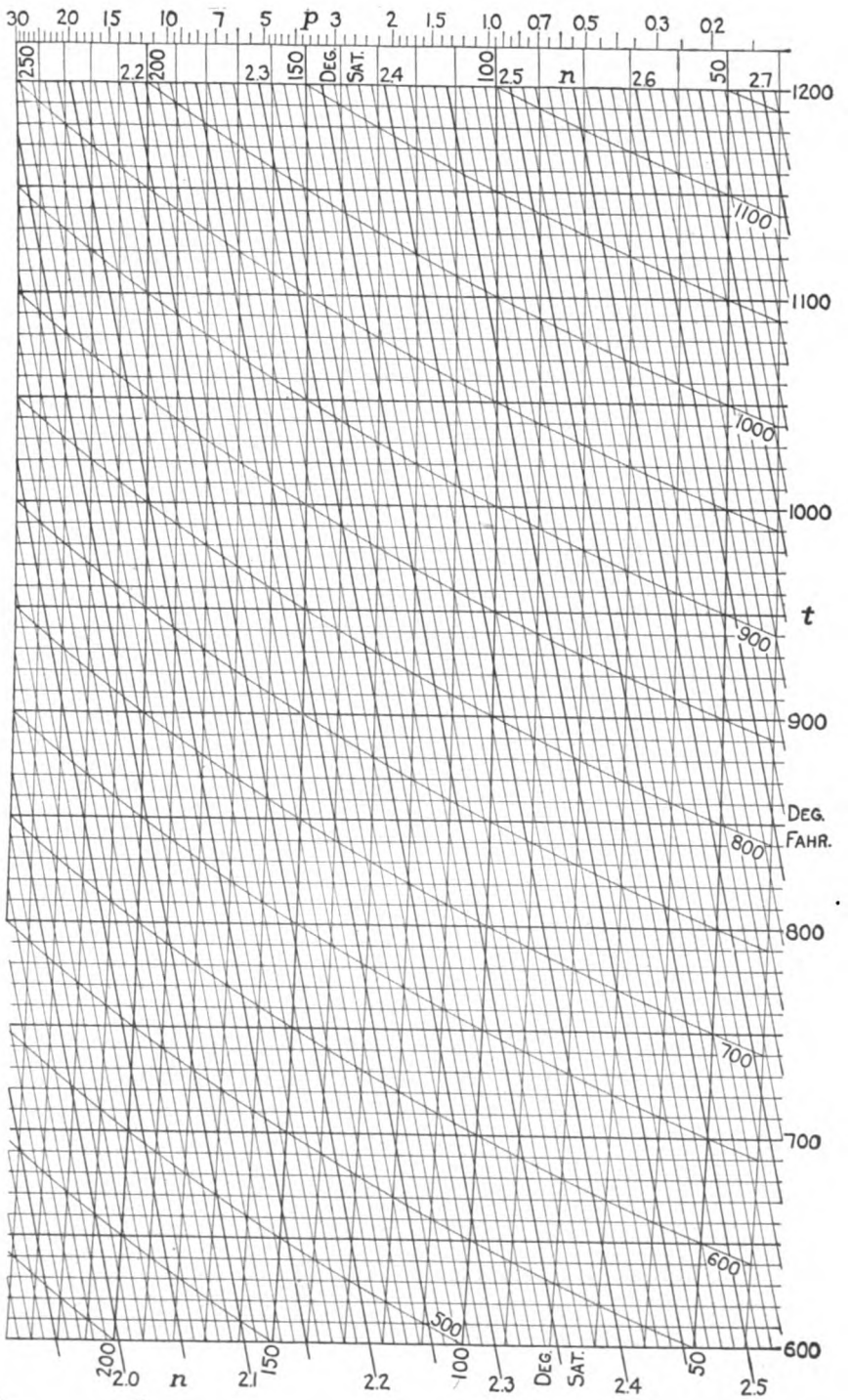

SUPERHEATED STEAM.

TABLE VIII, page E.

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(*From page* 579)

D. Notes on Steam-table Data

1. The preceding tables will be found to agree quite closely with the Marks and Davis *Tables of the Properties of Steam*. The differences lie within the range of variation of the better experimental data; but the changes which here appear are in direction of a smoother variation of the several quantities. The notes and references which follow do not attempt to cover the field of experimental investigation along this line; rather, they touch on a few important points, and may be considered as supplementing the " Discussion of Sources " by Marks and Davis.

2. The pressure-temperature relation follows the determination of Holborn and Henning, *Annalen der Physik*, 1908, Vol. 26, 833–883 — see also Heck, Trans. A. S. M. E., 1909, Vol. 30, 345–358. From 212 deg. to 400 deg. fahr. the values in the table are from the formula

$$\log \frac{p}{14.697} = 5.3807 \frac{t - 212}{T} - \frac{0.92563}{T} \frac{(689 - t)^4 - 477^4}{1{,}000{,}000{,}000}, \quad . \quad (a)$$

in which absolute temperature $T = t + 459.4$, corresponding with the commonly used centigrade value $T = t + 273$. This modification of Thiesen's formula agrees almost perfectly with the experiments. From 212 deg. down to 32 deg. the values given by Holborn and Henning are used directly. Above 400 deg. an interpolation was made among the data available in the year 1909; and up to 550 deg. fahr. this has been found to agree very closely with the authoritative Holborn-Baumann determination, *Annalen der Physik*, 1910, Vol. 31, 945–970: see also L. S. Marks, Jour. A. S. M. E., May 1911, and discussion by Author, September, 1911, page 1044.

3. The differential coefficient dp/dt was calculated from Eq. (*a*), and by the method of supplementary correction where the curve of p on t did not follow a formulated equation. This was for use in Clapeyron's equation,

$$\frac{r}{u} = \frac{144}{778} T \frac{dp}{dt}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (b)$$

which connects volume u with latent heat r.

4. The specific volume of steam (saturated and superheated) is in close accord with Linde's equation, of which the experimental foundation (see Fig. 43 ante) and the evaluation are given in *Zeitschrift des Vereines deutscher Ingenieure*, 1905, page 1697. His pressure function f_p is retained, but the temperature function f_t is adjusted so as to make s or u agree perfectly with r: above the range of Linde's formula (or above 200 deg. cent.) f_t departs more and more from his mathematical expression, which becomes of decidedly unsuitable value; and at very

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high temperatures it is very much of a guess, guided by the rational requirement that steam shall approach the law $pv = CT$ as it gets farther away from saturation.

5. As to specific heat of water and heat of the liquid, this table agrees with that of Marks and Davis in adopting the determination of Dieterici, *Annalen der Physik*, 1905, Vol. 16, 593–620. Above the experimental limit of about 600 deg. fahr. the curve in Fig. 45 is pure extrapolation, following what seems a probable course, so that the values of q are not to be considered at all authoritative.

6. Up to 400 deg. fahr. the latent heat r is guided by experimental values, mostly from electro-calorimetry. Of determinations at and below 212 deg., the most convenient summary will be found in an article by Prof. A. W. Smith, in *Physical Review*, 1907, Vol. 25, 145–170, or better, in *Monthly Weather Review* for Oct., 1907. Over the range from 212 to 400 deg., the later experiments of Henning are published in *Annalen der Physik*, 1909, Vol. 29, 441–465.

7. For both total heat H and latent heat r the derivation by Dr. H. N. Davis from the throttling experiments plotted in Fig. 68— see Trans. A. S. M. E., 1908, Vol. 30, 741–774 — is the real determinant, up to the limit of about 400 deg. fahr.* The departure from Davis' equation is called for and guided by the need of getting a smooth curve for the derivative dH/dt, which in his layout makes a slight but abrupt change from constancy to variability at 212 deg. The use of a smaller value of r at 212 deg. is based largely on the acceptance of Henning's determination at the value which he gives, without the increase of about 1.2 B.t.u. which Smith makes because of a supposed difference in the fundamental electrical units of measurement — Henning stating that this correction is not called for.

In the high range, above 400 deg., r is carried up with u by Eq. (b); the extrapolation is rational in idea and smooth and consistent in form, and at 1000 lb. pressure is probably of the degree of correctness named in § C above.

8. The law for total heat of superheated steam represented by the simple isothermal curves in Table VII is new, or at least makes prominent an idea which is only implied in other discussions. This scheme of interrelation, combined with experimental data as to the specific heat c_p, is believed to give a much better determination than can be made by experiment alone. The principal data used are those of Holborn and Henning (*Ann. Phys.*, 1905, Vol. 18, 739) and of Knob-

*These throttling experiments have been published as follows: Grindley, *Phil. Trans. Royal Society*, 1900, Vol. 194 A, 1-36: Griessmann, *Zeit. Ver. d. Ing.*, 1903, Vol. 47 II, 1852, 1880: Peake, *Proceedings Royal Society*, 1905, Vol. A 76, 185.

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lauch and Jakob (*Zeit. Ver. d. Ing.*, 1907, 81, 121), together with a rational derivation of the initial c_p, at or against the saturation line — see the paper by H. N. Davis, above referred to.

9. The calculation of internal energy and of entropy are purely numerical operations, not involving any combination or interpretation of data.

10. It will be noted that nothing has been said of the great work of Regnault in determining the properties of steam, upon which were based all the steam tables in use up to a few years ago. The changes made by recent experiment are not very great, when expressed in percentages, but they are sufficient to cause the values obtained by Regnault to be superseded.

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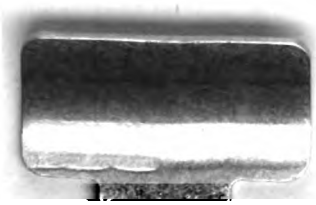

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